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ROYAL ENGINEERS INSTITUTE.

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EDITOR'S PREFACE.

THE issue of the present Volume (XVIII.) of the *Occasional Papers* for 1892 brings the publication up to date.

Of the five papers it contains, two are by officers of the Corps—Paper II., by Major R. H. Brown, treating of Egyptian Irrigation, and Paper IV., by Captain S. D. Cleeve, on the Application of Works to Irregular Ground.

There are two papers on electric science—Paper I., Electric Welding, by Graham Harris, Esq., M.I.C.E., and Paper III., Alternating Currents of Electricity, by G. Kapp, Esq.

Paper V. is a valuable treatise on Hydraulic Machinery, by G. Bodmer, Esq., A.M.I.C.E. It is very fully illustrated, and gives the latest information on the subject.

The latest tables of Service Ordnance, corrected to June, 1891, published by kind permission of the War Office, complete the Volume.

W. A. GALE, CAPTAIN, R.E.,
Secretary, R.E. Institute, and Editor.

November 1st, 1892.

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ERRATUM.

Plate XI., *Fig.* 4, "Hydraulic Machinery," Paper No. V. *For* "Plan of Dummy Lift," *read* "Plan of Dummy Piston."

PAPER I.

ELECTRIC WELDING.

A LECTURE DELIVERED

BY H. GRAHAM HARRIS, M. INST. C.E., M. INST. E.E.,

At the School of Military Engineering, on the 18th March, 1891.

ENGINEERS, as a rule, are not credited with having very vivid imaginations; it is not often that they have to exercise this faculty, and the result is that they are looked upon (and with truth) as prosaic individuals, incapable of realizing anything (except in connection with their profession) much in advance of that which they have directly before them or have been used to during their career.

This is due not to original want of imagination on the part of the engineer, but to the training which he undergoes, such training binding him down to strict facts and to accuracy of result, both of these being conditions antagonistic to the exercise of vivid imagination. In spite of all this I want you, for a minute or two, to exercise your imaginations, and to try and realize what would be the form of, say, an ordinary steam engine, or a traction engine, or "Steam Sapper," such as we have at work outside, if, instead of depending as we hitherto have done upon the mode of connection of the various parts of that machine by bolts and nuts, or (to a very limited extent, and with only one of the metals used) by welding, the engineer had at his command a metal or material capable of being readily shaped in any way he desired, as cast-iron or gun-metal is, and capable of being easily and readily united into a homogenous whole, as wrought-iron or steel can be, so that at the point of junction the strength obtained was as great as that of an equal area of the solid metal. If you will imagine this, I am sure you will agree that the present forms of machines have, to a large extent—and of necessity—resulted from the facilities which the engineer, when designing or constructing them, has had at his command.

You can well realize that were these facilities greater, especially in the direction of uniting or shaping the various pieces of various metals of which the machine is composed, the resultant machine would assume a very different and, we hope, a superior form. There is no doubt, also, that you will agree with me when I say that the uniting of two pieces of wrought-iron or mild steel by the primitive tools employed in hand-welding is an operation which cramps and limits the engineer in his designs and work. Further, you will agree that a process which enables him to more certainly unite the weldable metal—wrought-iron or mild steel—and which also enables him to unite by welding almost every metal to similar metal, and so far as we at present know to dissimilar metal, is one well worthy of full investigation by all interested in the profession of engineering.

Let us consider in this light the machine before you, an electric-welding machine, which I hope to show you in work directly. Here, in this part, you have cast-iron used, because the qualities distinguishing cast-iron are required for this particular portion of the structure, that is to say, the necessary shape can only be obtained at a reasonable cost from some metal capable of readily being made into the form you see here. The strength of cast-iron is, as you know, relatively a limited one, and therefore a larger quantity of the metal has to be used in order to enable this part of the machine to stand the strains which will be put upon it in working. Here you have a piece of gun-metal, or brass, used because it also can be readily shaped by casting, not used for the larger parts of the machine because of its expense, but used here because the piece is small, and therefore the question of expense is not of great moment. Remember, however, these metals can neither of them be satisfactorily united after the pieces are once cast, except by bolts and nuts. Here, however, you have a piece of wrought-iron, or mild steel, used because its strength is great, and because it can and has been readily united by welding.

Let us now consider another engineering structure—a suspension bridge, where the links by which the bridge is hung from the chains are of wrought-iron, or, nowadays, of mild steel. Unless these are made with the unsatisfactory weld obtained by hand, or by the steam hammer, the links have to be rolled out of the solid. In years gone by, when this form of bridge was more frequently employed than it is now, and when it was less easy to satisfactorily weld these pieces of wrought-iron because the steam hammer was not in common use, special machines were devised to enable the links of such bridges to be rolled from the solid bar with their enlarged ends or eyes, through which the connecting pins pass, rolled with them.

Figs. 1 and 2 show the various shapes of the bar before and after rolling by this process, and a rough illustration of the shape of the rolls by which the bar was gradually transformed from the plain rectangular block (shown at the top of *Fig.* 1) to the finished link shown at the bottom of that figure.

This mode of making suspension bridge links was in use nearly half a century since, that is to say, as long ago as 1845, by the firm of Howard & Ravenhill, of the King and Queen Ironworks, Rotherhithe. The successors of this firm were some of our most noted marine engine builders, even to very recent years.

As I have said, all this trouble was taken in order to avoid the necessity for a weld, because welding by hand, or even by the steam hammer, or by pressure from rolls, was, and is, unsatisfactory from its uncertainty. You can, of course, without my labouring the point, see that neither on the score of expense, nor on that of producing satisfactory work, would so good a job be made if a bar having originally and throughout its length the width it has at the eye were slotted down at the sides so as to bring it to the necessary width for the centre of the link; and, further, in bridge construction it is not desirable to leave this useless metal upon the sides of the bars, because of the extra weight thus imposed on the structure.

To return to our welding machine, and to the various sorts of metals of which it is composed, and to the existing modes of uniting these metals. You will realize that even an improvement in the certainty of connection of the one weldable metal commonly dealt with by engineers would be an improvement of the utmost use, and one the development of which would of itself create new uses and new fields.

Up to the present I have only dealt with the generalities of the question of uniting metals, whether by bolts and nuts or by welding.

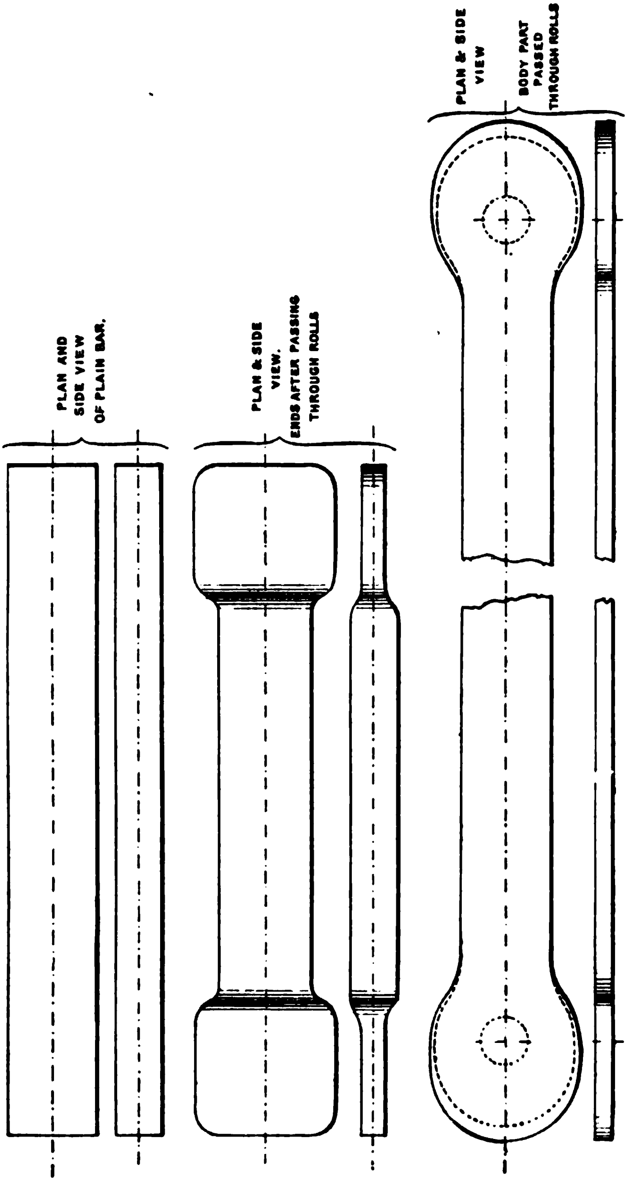
PLAN AND SIDE VIEW OF PLAIN BAR.
PLAN & SIDE VIEW. ENDS AFTER PASSING THROUGH ROLLS
PLAN & SIDE VIEW
BODY PART PASSED THROUGH ROLLS

FIG. 1.—*Suspension Bridge Links in Various Stages of Manufacture on Howard's Principle.*

Let us now see what peculiar quality is possessed by wrought-iron or by mild steel (which is practically only a clean form of wrought-iron) that enables this metal to be welded.

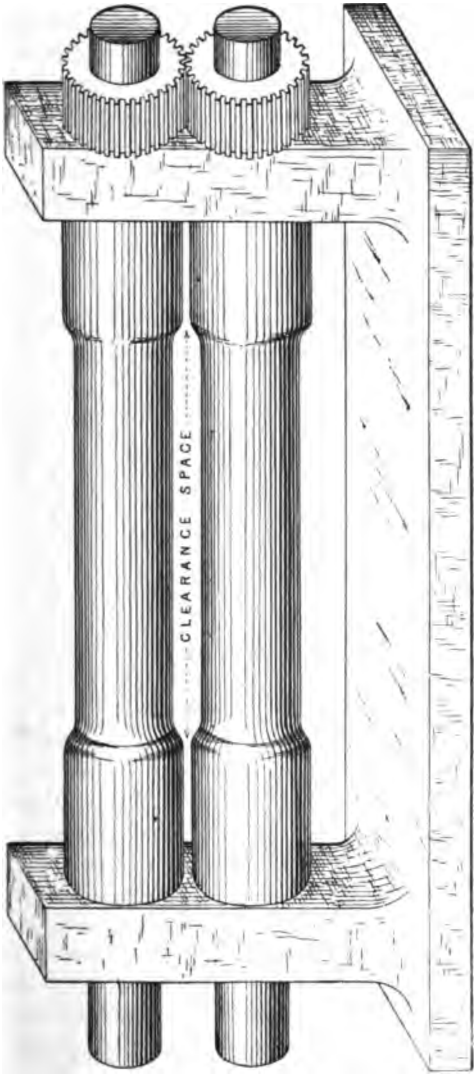

Fig. 2.—*Rough Sketch of Shape of Howard's Rolls for Rolling his Suspension Bridge Links.*

To do this we must at once define what is meant by true welding.

You know, most of you, pieces of cast-iron can be burned or fused together; you know that the same operation can be performed (perhaps more satisfactorily) with gun-metal; you know also that a common way of uniting gun-metal and similar alloys of metals is by soldering or brazing. Neither of these modes are really and truly welding. Welding consists in uniting pieces of similar or dissimilar metals in such a way as to cause the junction to become practically one with the piece upon each side of it, making the welded piece into a homogenous whole, and rendering the weld practically as strong as an equal area of the solid unwelded metal.

Although this may be considered a definition of welding, yet it does not state the conditions or qualities inherent in particular metals, which enable them to be united in this way, and I do not know that I can better impress upon you what true welding is, than by quoting to you that which Dr. Percy, in his book on the metallurgy of iron and steel (published as long ago as 1864), gives as his definition of welding. He says:—"Iron has one remarkable and very important property, namely, that of continuing soft and more or less pasty through a considerable range of temperature below its melting point. It is sufficiently soft at a bright red heat to admit of being forged with facility, as everyone knows, and at about white heat it is so pasty that when two pieces at this temperature are pressed together, they unite intimately and firmly. This is what occurs in the common process of welding. Generally, metals seem to pass quickly from the solid to the liquid state, and so far from being pasty and cohesive at the temperature of incipient fusion, they are extremely brittle, and in some cases easily pulverizable. But, admitting that there is a particular temperature at which a metal becomes pasty, its range is so limited in the case of the common metals that it would scarcely be possible to hit upon it with any certainty in practice, or, if it were possible, its duration would be too short for the performance of the necessary manipulation in welding."

You will see that the point upon which he insists in this definition is that the metals shall continue more or less pasty through a considerable range of temperature, in other words, shall remain in the pasty condition sufficiently long to enable the necessary junction to be effected, and this although subjected to the cooling effect of the surrounding atmosphere. He, of course, at that time had not in his mind the possibility of electric welding, where, as you will see presently, the heat is maintained or increased even while the weld is being made.

Although you have doubtless all of you seen the operation of welding by hand, yet I think it will not be wasted time if I show you this now; you will then more readily comprehend the differences between that mode of welding and the electrical mode. I do not know anything now left to engineers which illustrates more forcibly the skill of eye and hand to which an efficient workman can attain than the act of welding. I know of no more interesting sight than to watch this operation being performed, to see how the skilful smith will regulate (although he has the most crude and primitive implements to deal with) the temperature of the metal, the mode in which the junction is made, how he will ensure that the surfaces, when brought into contact, are clean and free from scale and foreign matter.

Let me further illustrate the difficulties of welding to you, so as to bring these more forcibly home to your minds, and to emphasize the necessary conditions which must be fulfilled in order to obtain a thoroughly satisfactory weld. To do this I will ask you to consider with me a series of welds such as were employed in uniting the tyres of the wheels of railway carriages and locomotives. I am glad to say it is many years ago since welds were discarded (entirely I hope) from the tyres of such wheels as these.

However carefully they were made, no matter what precautions were taken, a percentage of the welds were bad, and many very serious accidents resulted from the giving way of the tyres when the carriages were in motion at speed. So much was this the case that it was the common practice, when a weld was made in the tyre of a locomotive wheel, to put a mark upon that tyre upon each side of the weld, so that its position should never be lost sight of, and when any favourable opportunity presented itself, to inspect such weld. Further than that, means were devised whereby the tyre, even if broken, was retained upon the carriage wheel, with the object of enabling the carriage or locomotive to be brought to rest without leaving the rails, and without causing any very serious damage.

I will briefly describe this series of welds, and do it because they illustrate the successive precautions which experience showed to be necessary to lessen the percentage of failures. They will, I hope, impress upon you the conditions which, as I have said, must be fulfilled before a satisfactory weld can be made.

These welds are illustrated on *Figs.* 3 to 8. You will see on *Fig.* 3 the ordinary scarf weld; there the wheel tyre was cut at an angle, so that the junction between the two pieces might offer as much surface as possible, the hope being that the greater portion perhaps all, of that surface would be brought into effective union when the tyre was welded. On *Fig.* 4 you will see that the junction was made with a double scarf, this junction being called a "Bird's Mouth" weld. The difficulty in this weld was, that whatever dirt or foreign matter got between the surfaces, probably oxide, due to contact of the heated metal with the atmosphere, this foreign matter was not, with certainty, driven out in the act of welding, but the tendency was, owing to the shape of the surfaces, to retain it, and thus to render the weld unsatisfactory.

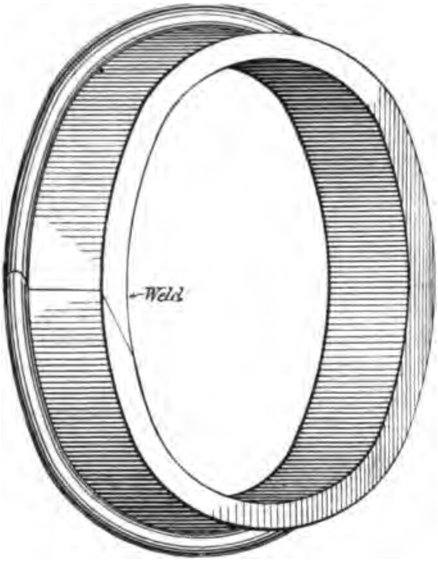

FIG. 3.—*Ordinary Scarf Weld.*

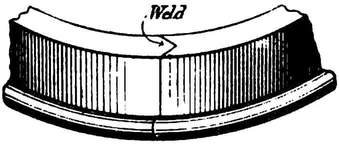

FIG. 4.—*Bird's-Mouth Weld.*

Fig. 5 shows another form of "Bird's Mouth" weld. In this the surface was increased. *Fig.* 6 gives that which was called the "single wedge" weld, where a piece shaped to suit an angular or wedge-shaped slot in the tyre was heated separately, and was forced into contact with the two cut surfaces of the tyre. This was a form of weld largely in use and practically with fairly satisfactory results. In *Fig.* 7 you see the "double wedge" weld, the surface in contact here being considerably increased.

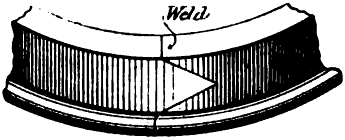

FIG. 5.—*Bird's-Mouth Weld.*

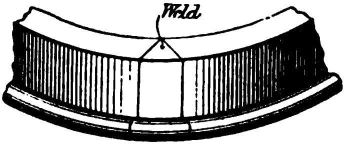

FIG. 6.—*Single Wedge Weld.*

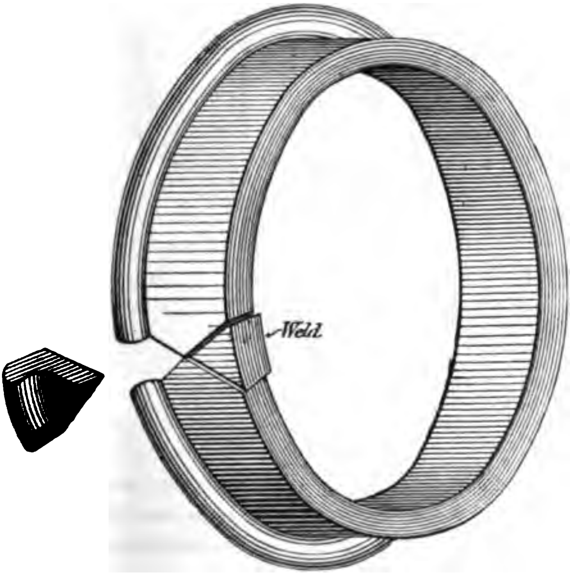

FIG. 7.—*Double Wedge Weld.*

All of these forms of weld were made by hammering, either by the sledge hammer in the hands of the workmen, or by the earlier forms of steam hammer or tilt hammer. But welds can also be made, and with equally, if not more satisfactory results, providing the metal is properly heated, by pressure, and you will see on *Fig.* 8 a welded tyre shown having a "brace" around it, and screws in this brace which, when screwed together upon the wheel tyre, pressed the heated surfaces into contact, thus excluding any "dirt," and, to this extent, ensuring the efficiency of the weld when made.

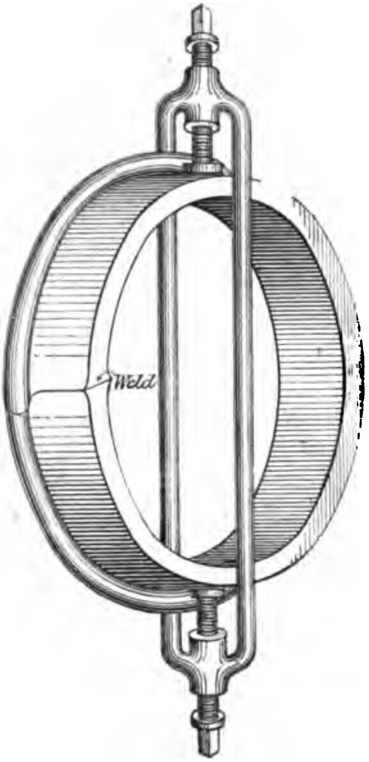

FIG. 8.— *Butt Weld.*

It was desirable, with such a weld, that the two surfaces should not be at an angle, the one with the other, and this form of weld was called a "butt" weld, and is the one which is used in electrical welding.

A mode by which the coils or tubes of heavy guns were at one time welded together is shown in *Fig.* 9.

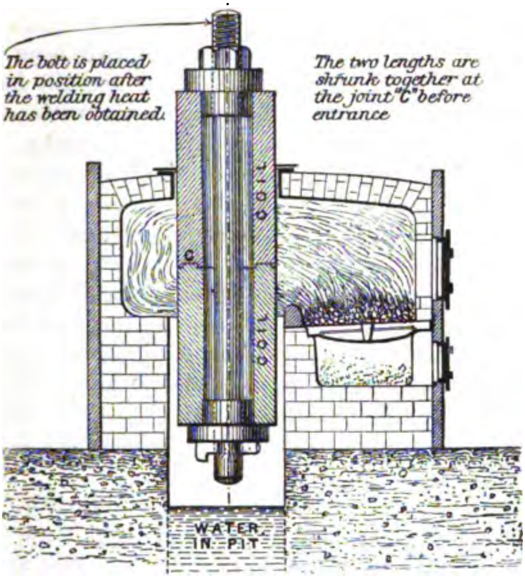
The bolt is placed in position after the welding heat has been obtained.
The two lengths are shrunk together at the joint "C" before entrance
COIL
C
COIL
WATER IN PIT

FIG. 9.—*Mode Employed in Early Days of Welding Big Gun Tubes.*

All the welds of which I have as yet spoken are welds in wrought-iron. Platinum is the one other metal which is equally capable of being welded by hand, because it will remain in the necessary pasty condition for a sufficient time, or through a sufficient range of temperature. I had at first intended to show you a piece of this metal being welded by hand, and in an ordinary fire, but time does not admit of my doing so; you may, however, assume that the operation is precisely similar to that of welding wrought-iron.

A little consideration will now enable us to realize what are the desirable conditions which should exist in order to ensure a satisfactory weld; these are: that the metal to be welded should be heated equally throughout its whole section; that the temperature should be capable of regulation, and, if possible, with the greatest nicety, and that this temperature should be maintained, or even increased, as the operation of welding is performed, and further, and this most important of all, that you should be absolutely certain no foreign matter, whether dirt from the fire, or oxide due to the exposure of the heated metal to the air, can, by any possibility, get between the surfaces which are to be united, and it is obvious that it is of the utmost advantage to be able to inspect the heating as it proceeds.

Unfortunately, in hand welding (and under that term I include welding under the steam hammer, or under the tilt hammer) it is only with the greatest difficulty the above conditions can be fulfilled. Thus welds are always matters of uncertainty, and it is mainly for this reason that wire ropes, which are without welds, are so rapidly superseding chains, where numerous welds are a necessity, in all cases where it is possible to make the substitution.

The heating in an ordinary fire must be from the outside inwards, and therefore the outside will be hotter than the inside, often being "burnt" before a welding heat has been obtained throughout. Further, it is impossible to inspect the condition of a piece being heated in an ordinary fire without removing it, and in this removal, and in the re-introduction of the piece into the fire, there is the possibility that "dirt" will be picked up, which will get between the surfaces.

Having premised this much as to welding as hitherto performed, and as to the conditions necessary for a successful weld, let us consider how far the desirable conditions just enumerated are possible, or can be obtained in electric welding.

You are, most of you, or perhaps I may say all of you, cognisant of the fact that the passage of a current of electricity through any material generates heat, and you are all of you doubtless aware that the amount of heat generated in any given conductor depends almost entirely upon the quantity of the current which is passing, and does not depend upon the electrical pressure of that current, or, as it is technically called, the electromotive force. You are also aware that the instruments by which the electrical "quantity" and the electrical "pressure" are measured, are instruments known respectively as the ammeter and the voltmeter. We have samples of these here upon the table.

In stating broadly, as I have done, that the heating effect of electricity depends upon the quantity of current, and not upon its pressure, I have purposely left out of consideration for the present a question with which we shall have to deal, and that is, the question of the difference of the heating effect produced by the same quantity of current passing through different conductors, and I also leave out of consideration for the present the other important question, that of the increase in the heating effect, due to the increase of temperature in the conductor through which the current is passing.

Let me show you an experiment in order to impress upon your minds the broad fact I have stated above.

We have here upon the table four lamps, ordinary incandescent lamps, in which, as you know, the light is produced by the passage of an electric current through a filament or thread of carbon enclosed in a glass bulb, from which all air has practically been completely exhausted.

The current to these lamps is controlled by a switch I have here, which, as you will see, is so arranged that I can turn the whole of the electricity through either one, two, or all the lamps. When I do this, the pressure in each, or rather the fall of pressure in each, will be the same, because they are coupled in series (as the electrician terms it), that is to say, the current is passing through the lamps in succession, and as the resistance to the passage of this current in each lamp is as nearly as possible the same, the amount of electrical energy being dissipated in the form of heat in each one is the same. You will see the heat is not very great, as the lamps do not glow very brightly, but I have purposely put only a small current through them. If I now turn the current on to three of them only, you will see that there is an increase in the heating effect, for the light given by each is increased. Please do not think that I am telling you that the ratio of the increase of heat and of light are the same; they are not. I am only telling you that the light has increased, because the heat has increased. The current is now only passing through three of them—the same electricity as before—but the total drop of pressure is now divided among the three, instead of the four, that is, in each is four-thirds of that which it was when the four lamps were in circuit. I will now turn it on to two of the lamps, when you will see that there is again an increase in light; and now I will turn it on to one alone, when you will see a further increase. Now the whole of the original energy of the electricity which was, when the four lamps were in series, being dissipated in the whole of them is dissipated in the one, and the lighting effect has greatly increased, because the heating effect has greatly increased.

I do not want to weary you, but I must tell you that this increase is not in the ratio of 4 to 1, as you would at first imagine, but is in the ratio of 16 to 1, and this because the length of the conductor, in this case the filament of carbon, of the lamp through which the electricity has been passed is only one-fourth of the length of the filaments in the four lamps; the dissipation of energy per unit of length is four times, and therefore the heating effect is four times four, or sixteen times that which it was when the four lamps were in series.

Remember that this heating effect increases or decreases almost in the ratio of the increase or decrease of the quantity of electricity passing.

This experiment will also have illustrated to you another fact, that the passage of a current through any material dissipates electrical energy (this being composed of quantity of current multiplied by pressure of current), and also that the energy dissipated in each lamp may be represented absolutely by the heating effect produced.

Let us now consider the first of the other points which I have reserved, that the heating effect of a current varies in different metals. I daresay most of you have seen the experiment which I am now about to show, where a chain or wire composed of different metals is so arranged that an electric current can be passed through it, and where the difference of heating effect in the different metals of which the chain is composed is visible to the eye, because of the difference of the light emitted. This chain is composed of alternate lengths of equal diameter silver and platinum wire, and you will see, when I pass the current through it, how the platinum lengths will be heated while the silver are not affected.

Now for the second point reserved, *i.e.*, that the heating effect increases as the temperature increases. I am afraid no simple experiment will enable me to illustrate this to you, and I must ask you to look at *Fig.* 10, showing by a curved line the increase of resistance to the passage of the current, and therefore the increase of heating effect due to an increase of temperature in soft iron wire.

This and the other diagrams (*Figs.* 11 and 12), which, however, I will not stop to describe, were prepared as records of extremely careful researches made by Dr. Hopkinson, the last President of the Institution of Electrical Engineers, and one of our most able electricians. They were published by him in the Philosophical Transactions for the year 1889, page 462.

Take this one (*Fig.* 10), which shows the increase for soft iron wire. The horizontal line at the bottom of the diagram shows the "line of unit resistance." Remember that "resistance" (which is the electrical term) may with any given piece of material be really represented by the heating effect due to the passage of a current through that piece.

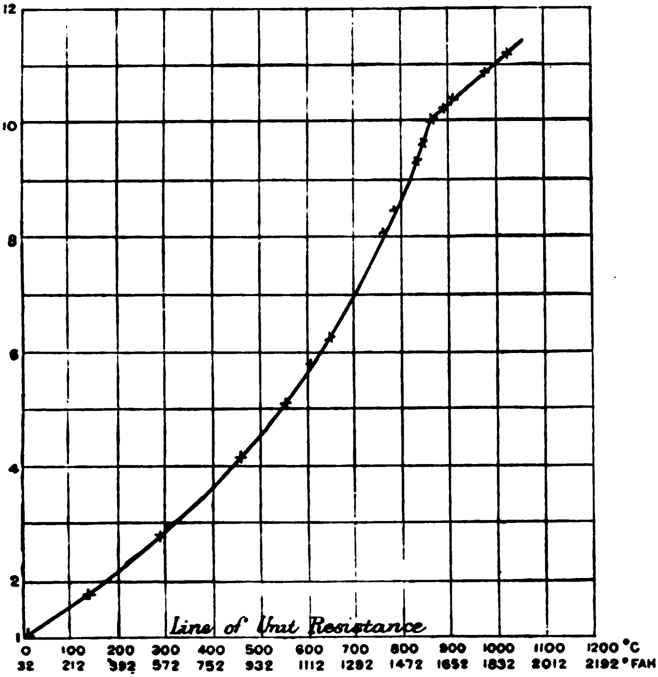

Fig. 10.—*Diagram of Increase of Resistance due to Increase of Temperature of Soft Iron Wire (Hopkinson, "Phil. Trans.,* 1889," *page* 462, *Plate* 19*).*

Let me now show you, in the welding machine itself, an experiment which, to my mind, illustrates very forcibly the whole of these three points which govern the heating effect of an electrical current. I have put into the jaws of the electric welding machine two hoops, or loops, of similar iron wire, each wire having exactly the same sectional area, but the hoops having different lengths, the one being twice the length of the other. As these are both secured at their ends in the opposite jaws of the machine, they are subject to the same sources of electrical current, and of dissipation of current, and it follows that there must of necessity be an equal electrical pressure at the similar ends of each of the hoops, that is to say, at the ends of the loops the pressure is the same. The hoops form the conductors for the electricity from one jaw of the machine to the other, and in these hoops the electrical energy will be dissipated.

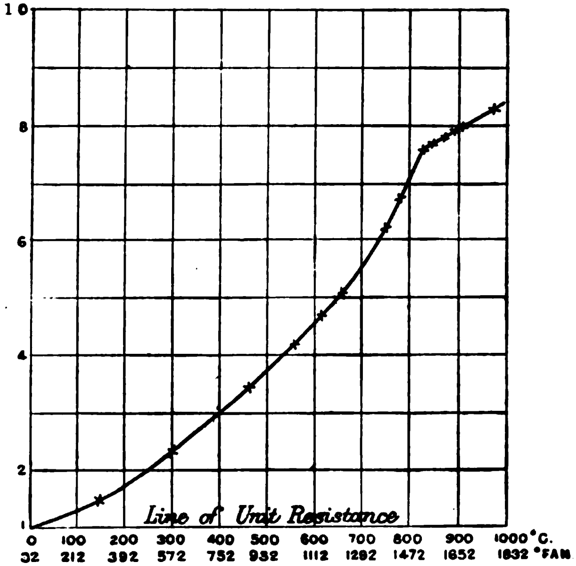

FIG. 11.—*Diagram of Increase of Resistance due to Increase of Temperature in Pianoforte Wire (Hopkinson, "Phil. Trans.,* 1889," *page* 462, *Plate* 19*).*

We will now have the current turned on, and the lights turned down, and will watch the heating effect.

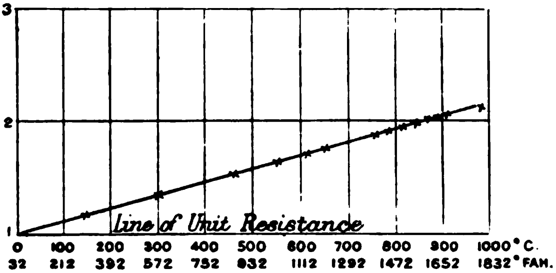

FIG. 12.—*Diagram of Increase of Resistance due to Increase of Temperature in Manganese Steel Wire (Hopkinson, "Phil. Trans.,* 1889," *page* 462, *Plate* 19*).*

You see that the shorter of the hoops is gradually beginning to glow, and it will, as a fact, glow more brightly than the long hoop, because the amount of electrical energy which is being dissipated in this hoop, per unit of length, is greater than is the energy being dissipated per unit of length in the longer hoop. Remember, the quantity of the current and the pressure of the current at these two ends of the hoops are the same, and the quantity and the pressure at those two ends of the hoop are the same, and therefore the energy dissipated per unit of length must of necessity be greater in the shorter hoop.

I want you, if you can, to watch the cumulative effect of continuing the passage of the current.

The shorter hoop gets hotter, the resistance increases as the temperature increases (and as we found, by the diagrams of Dr. Hopkinson's experiments, it would increase), with the result that an increased quantity of electricity passes through the longer hoop, thus increasing the rapidity of its heating.

I suppose it is impossible for you to accurately gauge, by merely looking on, that the heating effect is cumulative in its increase, but I must ask you to try to do so, and if you cannot, to take it from me as a fact.

You will, of course, see from all that I have told you that if we had a perfect conductor of electricity, no heating effect would arise, and that if we had a material which was absolutely impervious to electricity, and through which therefore no current could pass, no heating effect would arise.

Whilst I have been talking, the experiment with the hoops has been going on, and the shorter one has reached the point where no increase of temperature takes place, because the amount of current passing through it generates only sufficient heat to maintain the losses of temperature due to radiation of the heat from the surface into the atmosphere.

You will notice that I have again had to distinguish between heat and temperature, and I will just remind you here that it is a distinction of the utmost importance, when considering any heat questions. This I had to point out to you, in this room, when dealing last year with the question of "Petroleum as a Producer of Energy."

I think I have now shown you experiments enough to satisfy you that the heating effect of an electric current is dependent, in similar conductors of equal area, upon the dissipation of electrical energy per unit of length.

It will be well if you will remember these two, or rather three, points.

1. The variation of heating effect, per unit of length, produced in varying conductors of equal area, depends upon the material or metal of which the conductor is composed; this was illustrated by the Faraday chain of varying metals.

2. The increase of heating effect in any conductor increases as the temperature increases, no matter of what material or metal it is composed; this we learnt from Dr. Hopkinson's diagrams, and—

3. That the metals commonly to be welded by the engineer, wrought-iron or mild steel, are extremely well-placed, as regards the heating effect produced in them by the passage of a current, and as regards the increased heating effect in them due to increased temperature.

If you will remember these three points, I think you will have grasped the main facts in connection with the theory of electric welding by the Thomson system.

There is one other electrical question which I must deal with, and that is the question of the pressure and quantity of current needed to rapidly generate a welding heat in any given size of conductor of any given material.

I have shown you that the heating effect depends upon the quantity of the current, and does not to any great extent depend upon the pressure of that current. This being so, the question arises in practical work, what is the best and most convenient mode of producing an electrical current of great quantity and of low pressure?

Now, in order that the conductors bringing the current to the welding machine, where this current is to be used, should not themselves be heated, it would be necessary, if we only produced the current in the form of large quantity and of small pressure, and if the machines were far apart, to have these conductors of very ample area and very massive. It is obviously desirable, therefore, if such a current is to be produced, that the dynamo and the welding machine should be in very close proximity the one to the other, or, still better, all in one machine.

The earliest of the Thomson welding machines were made in this way, and there are photographs of them upon the table; but if you are dealing with a factory, where, probably, more than one welding machine is required, practical difficulties arise if you must have the dynamo and the welding machine in one; and, further than that, a dynamo, to produce current in great quantity and of low pressure, is a machine much more massive and cumbersome than one to produce current of high pressure and of small quantity. It is obvious, therefore, that if by any means we can transform the current of high pressure and of small quantity produced by the smaller dynamo into the current of low pressure and large quantity needed in the electric welding machine, it would be advantageous, especially if, as you will have gathered will be the case, small conductors from the dynamo to the machine may be used to carry the small quantity of current, because it is of high pressure. Under these circumstances it is possible to use the more economical and less cumbersome dynamo, not only to drive one welding machine, but, by dividing up the current produced, to drive many electric welding machines in various parts of a factory. Well, we can transform the current, or rather, we can transform a high pressure alternating current into a low pressure continuous current.

This transformation is effected by an apparatus practically the reverse of that which is known as the Ruhmkorff, or "Induction," coil.

In the Ruhmkorff coil there is a comparatively short length of wire of large diameter, through which the current, say from a battery, is put, this wire being wound around a core of soft iron, on which there is a coil of very fine wire of much greater length than that of the large wire, which is called the primary circuit, the fine wire being called the secondary circuit.

I have here upon the table a Ruhmkorff coil, or rather two, kindly lent to me by Professor Dewar, of the Royal Institution, and I may tell you that this is one of the coils used by Faraday in his experiments on high potential electricity.

I will just give you a brief illustration of the possibility of this transformation of quantity and pressure of the electric current into pressure and quantity of electric current, but we must not spend much time upon it.

The first Ruhmkorff coil is connected to a battery of cells below the table which are delivering their current into the primary circuit of this coil, and you will see the spark which I am able to get at the terminals of the wire, because the current is converted by the coil from the low pressure, or potential, of the cells into the high potential current capable of leaping across this space, this spark coming from the secondary (the fine wire) of the coil. Now, if we connect the secondary of this coil to the secondary of the other coil here, we shall produce in the primary of the second coil a current practically of the same character, less the losses of conversion, as that originally given by the five cells, that is to say, we shall have first transformed the low pressure current produced by the battery into the high pressure current delivered by the secondary of the first induction coil, capable, as you will see, of giving a spark of the intensity which I am now showing you, and we shall then re-transform through the second coil the current thus produced into practically, with its original pressure, the original current.

I have hung in the circuits incandescent electric lamps, and you will see that where the quantity is great, and the pressure or intensity low, those lamps are heated, and emit light, but where, although the same electricity is passing, the intensity is great, and the quantity is low, similar lamps are not heated, and no light is emitted from them.

Now, as I have told you, in the Thomson electric welding plant, the dynamo which is used produces the current, an alternating one, in the form of high pressure and of small quantity; it is shown in *Figs.* 13 and 14. That current is being generated by the dynamo placed in the shed outside the building (which all of you will be able to see to-morrow), the dynamo being driven by one of your own "Steam Sappers." Something like 18 indicated horse-power is being put into the dynamo. The current from this dynamo is being brought through the small diameter cable you see here, then through a resistance coil (the action and details of which I will presently describe to you), to the machine, or rather to the transformer, the reversed Ruhmkorff coil, which is a part of the welding machine.

The transformer is shown in longitudinal section on *Fig.* 15, and in cross section on *Fig.* 16, and those of you who know the details of the Ruhmkorff coil will, when I explain this machine to you, readily recognize the likeness between the two. Look at this longitudinal section. Here you have a series of discs of soft sheet-iron, insulated the one from the other by brown paper placed between them, and held tightly together by these through bolts, thus making a hollow cylinder, or sleeve, composed of these wrought-iron discs. Round about the walls of this cylinder is wound, inside and out, a length of insulated copper wire. I am told that in the machine here before us there are altogether 76 convolutions of this wire wound lengthways round the coil.

Centrally through the hollow left in the centre of the coil there is this hollow copper core, or cylinder, and to the ends of this core are attached the main copper conductors which are in electrical connection with the jaws carrying the pieces to be welded.

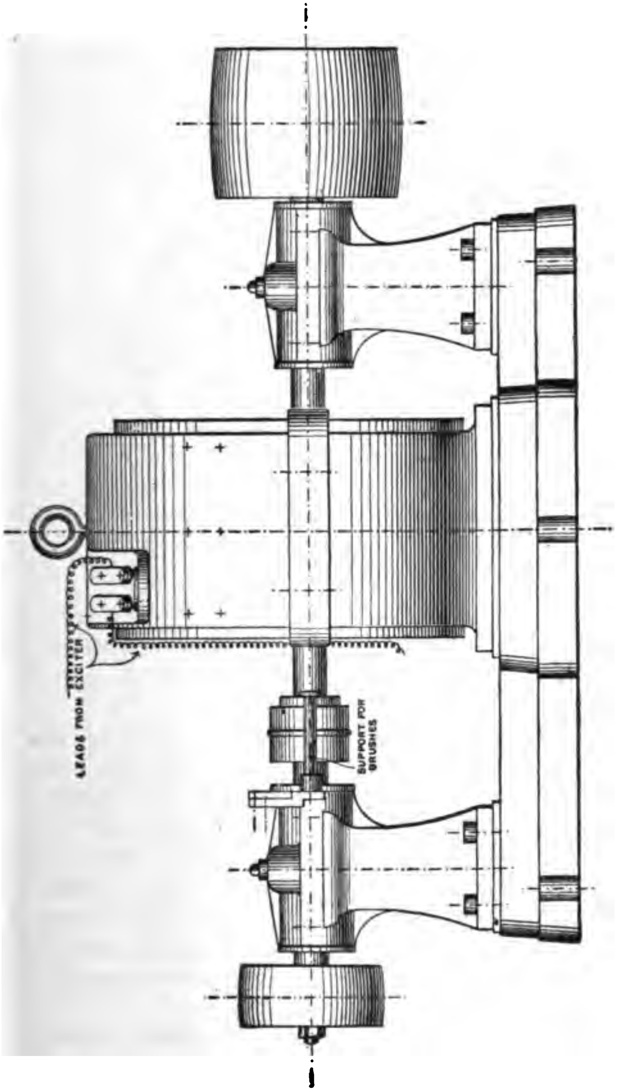

FIG. 13.—*Front Elevation of Thomson-Houston Alternating Current Dynamo used with Large Welder.*

The high pressure current from the dynamo passes into this external wire, which is wound around the discs, and is transformed by this combination of parts into the current of great quantity and low pressure needed for the welding.

Now let us consider the action of the machine itself; it is shown on *Figs.* 17 and 18.

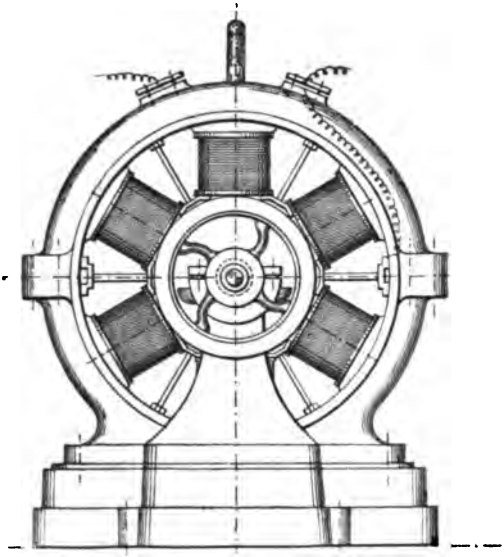

FIG. 14.—*End Elevation of Thomson-Houston Alternating Current Dynamo used with Large Welder.*

You will see that the gun-metal jaws which hold the pieces to be welded are capable of being moved the one towards the other, or rather that one is fixed and the other is capable of being moved towards it. You will understand the operation of the machine much better if I show you a weld being made. But, before doing this, there is one further piece of the apparatus which I must explain, and that is the resistance coil, to which I have already referred. By its means the amount of electricity passing through the pieces to be welded, and thus the heat generated in these pieces, can be regulated with the utmost nicety.

This is the implement itself here, and it is shown in *Figs.* 19 to 21.

It consists of a light vertical cylindrical case, having slate ends—slate being a non-conductor of electricity. In this case there are a number of vertical German silver wire coils, German silver being used because the heating effect of a current upon it is extremely slight. These coils pass through the slate insulation at the top, and terminate each in a gun-metal block, the blocks being insulated the one from the other, and the whole forming a ring, with alternate insulators and gun-metal blocks. A central wooden spindle carries upon it an arm, terminating at its extremity in a split piece of brass,

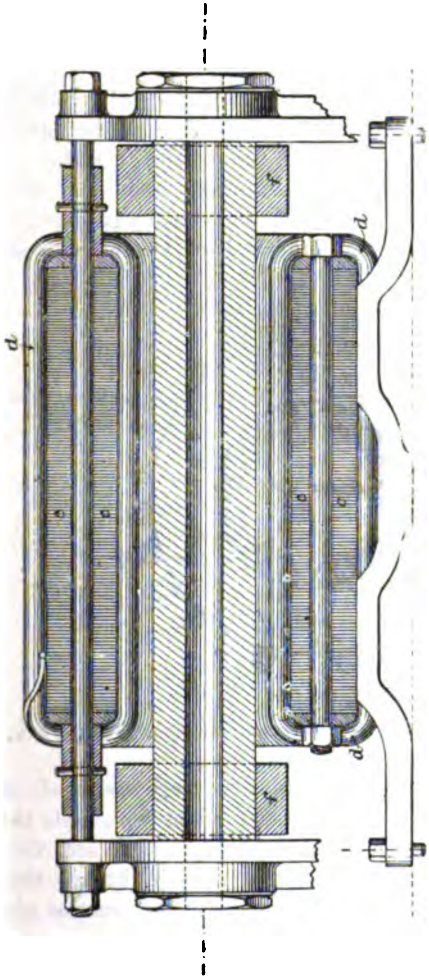

Fig. 15.—*Longitudinal Section through Transformer.*

so arranged that it will always tend to keep in contact with the top surface of the ring, formed partly of the metal blocks and partly of the insulation between these. The top part of this split piece is in contact with the underside of a gun-metal ring, and the whole arrangement is such that as the arm, or contact bar, having the split gun-metal piece upon it is moved in one direction, the German silver wire coils are, in succession, brought into the circuit from the dynamo to the transformer of the welding machine. As this bar is moved in the opposite direction, these German silver wire coils are, in succession, taken out of the circuit. In this way the current from the dynamo to the transformer of the welding machine can be readily caused to pass through a greater, or less, number of these coils, each of which offers a certain resistance to its passage, there is thus delivered to the machine a greater or less amount of electricity, because more or less of it will have been absorbed, or dissipated, by the varying number of German silver wire coils through which it has passed. By these means, as you will see, the operator at the machine regulates the current passing through the pieces to be united, and is thus able to govern with the greatest nicety the heat at the point of junction.

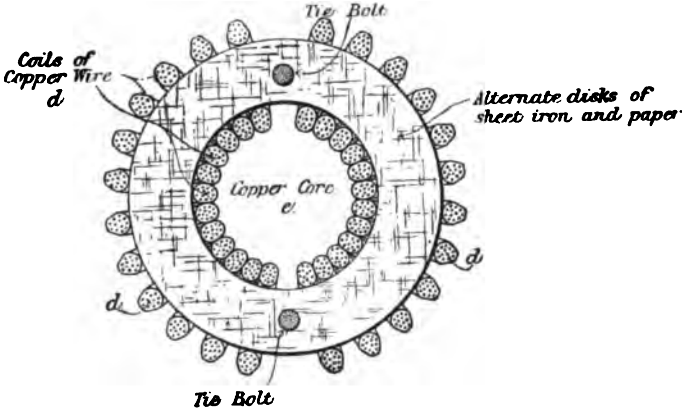

Fig. 16.—*Cross Section through Transformer.*

I am afraid that many of you will say that this machine must be extremely complicated, because of the difficulty which I have found in describing it, and the difficulty which I am afraid many of you also will have in understanding it (probably increased by that description) but I think I can show you that learning to work

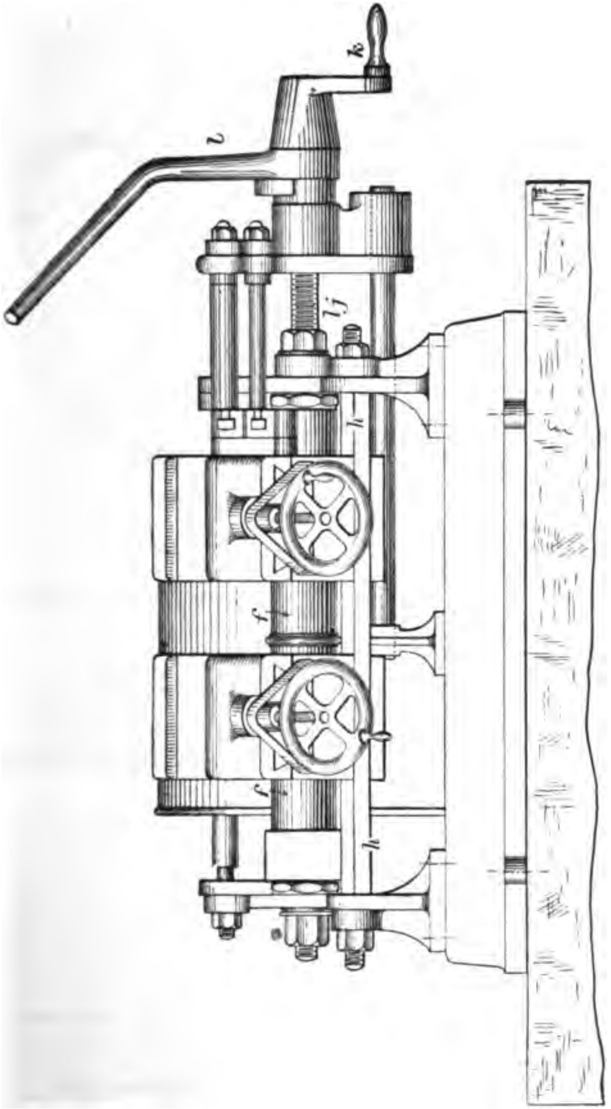

FIG. 17.—*Front Elevation of Large Welder.*

the machine is a much more simple operation than learning to understand it.

Two of your Corps—those who have made the hand-welds for us—have been taught by Mr. Pond here, the electrician, how to

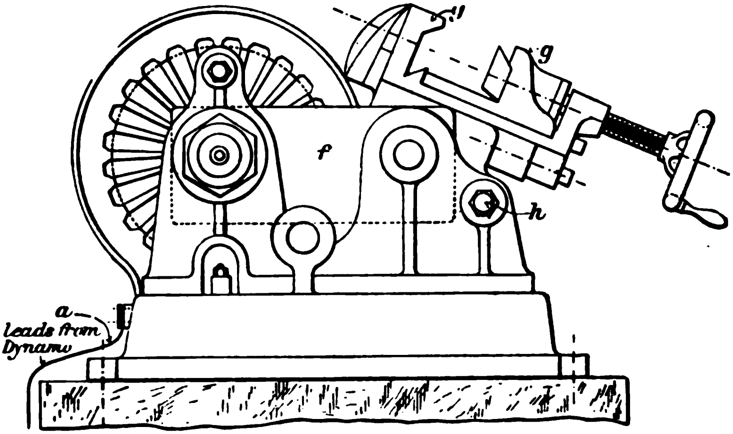

Fig. 18.—*End Elevation of Large Welder.*

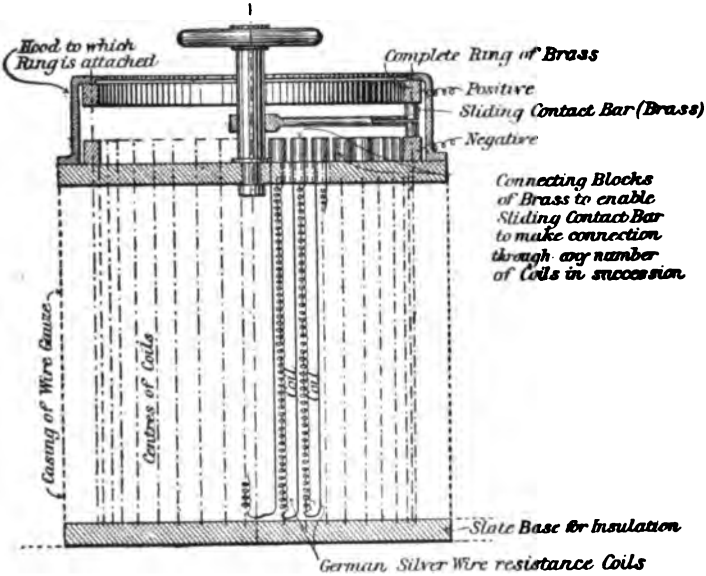
Hood to which Ring is attached
Complete Ring of Brass
Positive
Sliding Contact Bar (Brass)
Negative
Connecting Blocks of Brass to enable Sliding Contact Bar to make connection through any number of Coils in succession
Casing of Wire Gauze
Centres of Coils
Coil
Coil
Slate Base for Insulation
German Silver Wire resistance Coils

Fig. 19.—*Sectional Elevation through Resistance Box.*

make welds in the machine, and I will ask them to make a weld for you to see.

I may tell you that at the second attempt they made a very successful weld.

I daresay many of you will say: "This is apparently one mode of producing the heat electrically in metals to be united by welding. Surely there is another mode, and why should it not be adopted?"

Those of you who know the action of an electric arc light know that the light is produced by the extremely high temperature generated in the carbon points, due to the passage of the current between them, and you will say that if this temperature can be readily obtained, why should not the pieces of metal to be united be heated by the radiant heat of the electric arc?

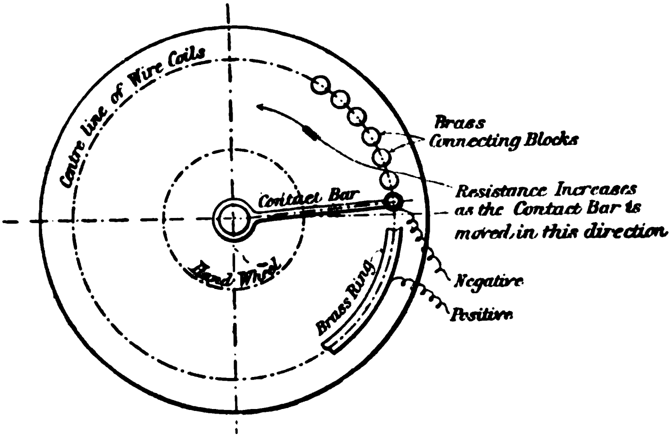
Centre line of Wire Coils
Brass Connecting Blocks
Contact Bar
Resistance Increases as the Contact Bar is moved in this direction
Hand Wheel
Brass Ring
Negative
Positive

FIG. 20.—*Plan of Resistance Box.*

I may tell you that this form of electric welding has been, and is, in use, but it cannot be—for many reasons which you will realize, if you consider the matter—as satisfactory in its operation in all respects as is the Thomson mode.

Time is getting on, and as I want to show you the machines in work on different metals and sections of metal, I will not delay to give these reasons, but will just call your attention to the fact that, in this second mode of welding, the heat is not produced directly in the piece to be welded, nor is there the certainty of keeping the surfaces to be united clean, as in the Thomson mode.

Let us now make another plain weld in the machine, and let us consider how the operation is performed.

You will see that the ends of the bars to be united are simply rough broken ends, the bar having been nicked round with a chisel,

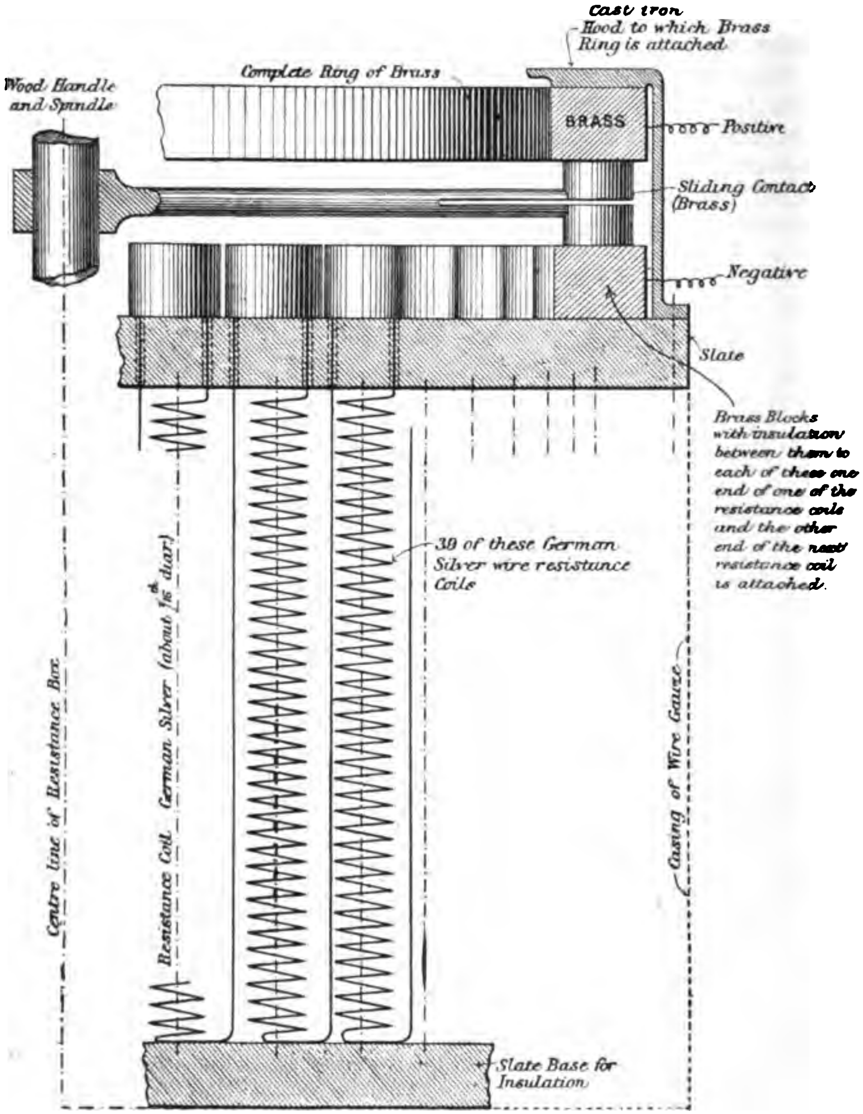
Cast iron
Hood to which Brass Ring is attached
Complete Ring of Brass
Wood Handle and Spindle
BRASS
Positive
Sliding Contact (Brass)
Negative
Slate
Brass Blocks with insulation between them to each of these one end of one of the resistance coils and the other end of the next resistance coil is attached.
39 of these German Silver wire resistance Coils
Centre line of Resistance Box
Resistance Coil. German Silver (about 1/16th diar)
Casing of Wire Gauze
Slate Base for Insulation

FIG. 21.—*Detail Section through Part of Resistance Box.*

and then broken. These pieces of bar are clamped into the jaws, as you see, and the rough broken end of one is brought into contact with the rough broken end of the other. The current is then switched on, and begins to pass through the points of contact between the rough ends of the bars. These points become heated, and therefore offer a greater resistance, because of their increased temperature, to the passage of the current. Pressure is applied by the screw and hand-wheel to force the bars into contact, and, as they become sufficiently heated, the metal of the points in contact flows away, thus allowing of other and cooler parts of the surface coming into contact. These are heated; their resistance is consequently increased; the current again seeks the cooler parts, and pressure being continuously applied, in a very little time the whole surfaces of the ends of the bars are sufficiently heated, and are forced into close union. The current is then switched off, the bar is taken from the machine and hammered on an anvil in the ordinary manner.

Now, let us consider how wonderfully natural causes have assisted us in this operation of electric welding, and how simple the whole matter is.

First, the bar ends which have to be united need not be prepared in any way whatever; whether rough or smooth, it matters not.

Next, it is only the points which are in contact, and from which, therefore, the air is excluded, which are first heated, and as the pressure is applied the heated metal flows on one side, and carries with it any oxide there may have been formed by contact of the heated metal with the air, or any dirt or foreign matter of any kind which would otherwise prevent close union.

Further, we have been able to examine closely the heat throughout the whole time of heating; we have been able to increase it, or to diminish it, at will; and

Lastly, and most important of all, the heating has taken place practically from the inside outwards.

As a fact—although I don't like to take away from the credit of those who have been using the machine—it is almost impossible, with ordinary care, to make an unsatisfactory electrical weld in these machines.

I have upon the wall records of very many experiments (which I superintended) upon a complete electric welding plant, in welding 1⅛-inch round iron bars (see *Fig.* 22 and the table at end). There are also copies of the indicator diagrams (see *Figs.* 23, 24, and 25) taken from the engine driving the electric welding machine, and a complete statement of the tests, both for tensile strength and for bending hot and cold, made at the Testing Works of Mr. Kirkaldy, in order to ascertain the strengths of the welds, and also

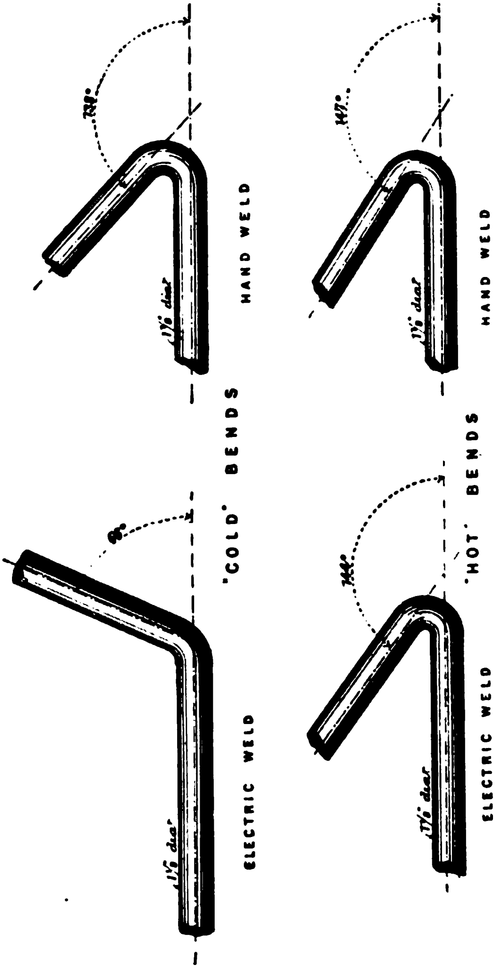

FIG. 22.—*Diagram showing Angle of Bend in Kirkaldy's Tests.*

a comparison of the strengths of a large number of welds made by hand in the ordinary way, and from the same bar-iron, by a very skilled smith, who was instructed to do the very best he could to beat the electric welding machine.

You will see that while the average strength of the bar per square inch of original sectional area was 52,642lbs., the average strength of the electrically welded bars, which broke at the weld, was 48,215lbs, or practically 95 per cent. of the original strength, while the average strength of those welds made by hand was 46,899lbs., or about 87 per cent.

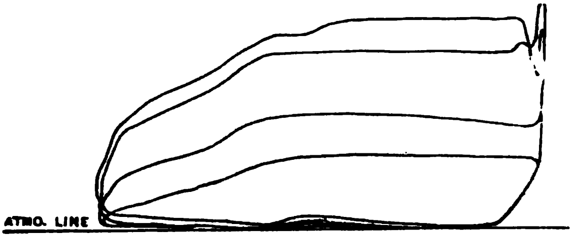

Fig. 23.—*Series of Four Diagrams taken during the Progress of a Weld.*

Boiler pressure, 83lbs. ; mean pressure, 30·59lbs. ; revolutions, 122 ; mean indicated HP., 37·16. Scale—60lbs. = 1 inch.

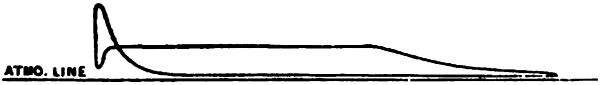

Fig. 24.—*Engine Running Light.*

Boiler pressure, 82lbs. ; mean pressure, 7·54lbs. ; revolutions, 128 ; indicated HP., 9·69. Scale—60lbs. = 1 inch.

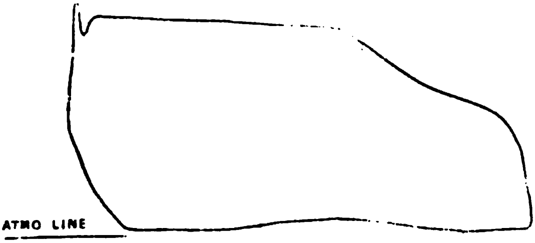

Fig. 25.—*Maximum Indicated HP. during a Weld.*

Boiler pressure, 79lbs. ; mean pressure, 53·45lbs. ; revolutions, 119 ; indicated HP., 50·78. Scale—60lbs. = 1 inch.

Remember, the electric welds are all "butt" welds, while the hand welds were all "scarf" welds.

In the early part of my lecture I pointed out to you—or I asked you to imagine—the possibilities of a process whereby similar metals, other than wrought-iron and platinum, would be united by welding, and of a process which should also admit of dissimilar metals being united in that way, and if you will remember Dr. Percy's definition of welding you will recognise that the difficulty with metals other than wrought-iron and platinum has been to keep them in the necessary pasty condition for the time requisite to allow of the junction being made; but, as I have shown you, electric welding enables you, while the operation is going on, to maintain, or even to increase or diminish, the temperature, and, as a fact, almost every metal can be united by means of these machines, and the combinations which have been made are too numerous for me to read out. Every variety of steel is weldable, every variety of iron, cast-iron, malleable iron, every variety of copper, of brass, in fact, a list of something like 80 or 90 different metals, or combinations of metals, which have been welded electrically is before us.

Aluminium is easily welded; bronze has been welded to iron; but there, I must not enumerate any more examples.

With all those metals which are difficult to weld, because, when a critical temperature is reached, they do not remain in the pasty condition, but immediately fuse, an automatic welding machine, as it is called, such as the small one we have here, is employed.

This is so arranged that a steady pressure being applied to the ends of the pieces to be united, and the current being turned on, the moment the proper temperature has been attained the pressure forces the pieces closer together, "upsetting" the heated ends, and this movement causes a switch to be operated, switching off the current from the machine.

I will ask Mr. Pond to weld a piece of copper wire in the automatic machine, so that you may see the operation.

I could give you a very long list of the uses to which these machines have already been put, but my time has almost elapsed, and I will confine myself to two.

Mr. Webb, the locomotive engineer of the North Western Railway, has had two of these machines in work at the Crewe Works, I think, for over a year, and he told me the other night that he had employed one of them almost entirely upon repair work, that is to say, a locomotive would come in with some part, say of the valve gear, broken, a small piece of metal, perhaps, but valuable because of the machine work which had been done upon it, in order to shape it for the purposes for which it was required. With such a piece it would be impossible to repair it by welding it together by hand, without injuring the piece so seriously as to render it necessary practically to re-machine it all over, but, with electric welding, it is only necessary to put it into the jaws of the machine, switch on the current, and the weld is made, and this without injuring the machined work in the very least. A file is put over the point of junction, and the piece, saved from the scrap heap, and equal to a new piece, is ready to go into a new locomotive.

One small item of which Mr. Webb told me was that in old locomotives, as they come in, the standards supporting the hand-rail around the foot-plates are, as you know, turned with a turned pin at their foot. This is rivetted through the plate or frame, and when the boiler is done with, these turned standards used to be thrown away, because, in order to get them out, the pins had to be cut off, and to hand-weld on a new pin would spoil the standard, that is to say, it would need to be re-turned.

Nowadays, he gets them out as he used to—by cutting off the pin—but he welds on a new turned pin electrically, thus making a standard equal to new, and effecting another saving due to the electric welding machine, and so on. He told me he could give a hundred other illustrations of uses to which he, in his business, is able to put the machines which he has.

There is one development of the electric welder which will, perhaps, interest soldiers more than any other, and that is the use which is made of it in America for manufacturing common shell for big guns, and it is suggested that the metal cases for fixed ammunition shall be similarly made.

In the case of the shell, a piece of steel tube, of the necessary thickness and diameter, is taken, and a length equal to the straight part of the shell is cut from this tube. A turned steel butt is then welded on to one end of this piece, and a hardened steel point is welded on to the other, and thus a steel shell is formed which is absolutely without the possibility of suffering, as cast-steel shells very frequently suffer, from flaws, causing internal strains, which result in the spontaneous breaking-up of the shell, and which may have its point, its butt, and the body part all of different metals, or of the different qualities of the same metal most suitable for the particular parts.

I will now ask Mr. Pond to make different sorts of welds for you to see, and, as I hope the machine will be left here to-morrow, I have no doubt you will all of you find some opportunity of inspecting the process, and those of you who care to do so, can learn to perform the operations themselves.

I cannot leave off talking without thanking you for the courteous way in which you have listened to me, and for the great attention you have paid to that which I am afraid I have wearied you by telling you.

RESULTS OF EXPERIMENTS TO ASCERTAIN THE TENSILE

NOMINAL SIZE, 1⅛-INCH DIAMETER.

Broke in Solid.

Test No.	Description.		Original.		Ultimate Stress.			Contraction of Area at Fracture.	Extension in 10 Inches.	Appearance of Fracture.
			Diameter.	Area.	Total.	Per Square Inch of Original Area.				
Y			Inch.	Sq. In.	Lbs.	Lbs.	Tons.	Per cent.	Per cent.	
405	F B 2	Electric Welded.	1·11	0·968	52,875	54,623		53·1	24·2	Fibrous.
407	,, 4		1·11	0·968	50,890	52,572		53·1	28·1	,,
409	,, 6		1·11	0·968	50,480	52,149		50·6	24·2	,,
411	B 8		1·11							
413	,, 10		1·11	0·968	52,410	54,143		49·4	22·7	,,
415	,, 12		1·11	0·968	49,840	51,488		50·6	20·6	,,
417	,, 14		1·11							
419	,, 16		1·11							
421	,, 18		1·11	0·968	50,910	52,592		51·8	23·9	,,
423	,, 20		1·11							
425	,, 22		1·11							
427	,, 24		1·11							
429	,, 26		1·11							
431	,, 28		1·11	0·968	51,335	53,032		51·8	20·8	,,
433	,, 30		1·11	0·968	50,260	51,921		49·4	19·1	,,
435	,, 32		1·11	0·968	50,060	51,715		49·4	20·6	,,
437	,, 34		1·11	0·968	50,745	52,423		51·8	21·2	,,
439	,, 36		1·13	1·003	51,710	51,555		52·3	23·9	,,
441	,, 38		1·11	0·968	51,645	53,352		49·4	21·1	,,
443	,, 40		1·13							
445	,, 42		1·11	0·968	50,665	52,340		49·4	20·9	,,
447	,, 44		1·11	0·968	52,260	53,988		50·6	22·2	,,
449	,, 46		1·11	0·968	50,195	51,854		49·4	22·0	,,
451	,, 48		1·11							
453	,, 50		1·11	0·968	51,280	52.975		49·4	20·6	,,
455	,, 52		1·12	0·985	52,555	53,355		50·3	21·8	,,
457	,, 54		1·11	0·968	51,480	53,182		48·0	20·4	,,
459	,, 56		1·11	0·968	50,385	52,051		50·6	22·1	,,
461	,, 58		1·11	0·968	51,320	53,017		49·4	21·7	,,
463	,, 60		1·11							
465	,, 62		1·11	0·968	50,065	51,720		49·4	21·2	,,
467	,, 64		1·11							
469	,, 66		1·11							
471	,, 68		1·11	0·968	51,355	53,053		53·1	21·6	,,
473	,, 70		1·12							
475	,, 72		1·11							
477	,, 74		1·12	0·985	51,310	52,091		48 9	22·6	,,
479	,, 76		1·12	0·985	51,155	51,934		51·5	23·4	,,
481	,, 78		1·12							
483	,, 80		1·12	0·985	52,245	53,041		52·7	23·6	,,
					Mean	52,646	=23·5	50·6	22·2	

STRENGTH, ETC., OF SIXTY-EIGHT ROUND IRON BARS, WELDED.

BRAND, FARNLEY.

Broke in Weld.					
Original.		Ultimate Stress.			Ratio of Weld to Solid.
Diameter.	Area.	Total.	Per Square Inch of Original Area.		
Inch.	Square Inch.	Lbs.	Lbs.	Tons.	Per cent.
1·12					
1·13					
1·11					
1·12	0·985	49,775	50,533		96·3
1·13					
1·15					
1·14	1·021	50,065	49,035		93·4
1·16	1·057	49,740	47,058		89·7
1·14					
1·13	1·003	42,485	42,358		80·7
1·13	1·003	49,315	49,167		93·7
1·13	1·003	44,740	44,606		85·0
1·13	1·003	49,705	49,556		94·4
1·15					
1·14					
1·16					
1·15					
1·14					
1·14					
1·14	1·021	49,185	48,173		91·8
1·14					
1·12					
1·15					
1·13	1·003	52,240	52,084		99·2
1·14					
1·15					
1·13					
1·14					
1·14					
1·14	1·021	48,045	47,057		89·7
1·12					
1·11	0·968	43,570	45,010		85·8
1·13	1·003	48,215	48,071		91·6
1·12					
1·12	0·985	48,380	49,117		93·6
1·12	0·985	48,745	49,487		94·3
1·14					
1·13					
1·13	1·003	52,070	51,914		98·9
1·13					
		Mean	48,215 =	21·5	91·9

Broke in Solid.

Test No.	Description.		Original. Dia-meter.	Original. Area.	Ultimate Stress. Total.	Ultimate Stress. Per Square Inch of Original Area. Lbs.	Ultimate Stress. Per Square Inch of Original Area. Tons.	Contraction of Area at Fracture.	Extension in 10 inches.	Appearance of Fracture.
Y			Inch.	Sq. In.	Lbs.	Lbs.	Tons.	Per cent.	Per cent.	
639	B 92	Hand Welded.	1·11							
641	,, 94		1·11							
643	,, 96		1·11							
645	,, 98		1·11							
647	,, 100		1·11							
649	,, 102		1·11	0·968	50,360	52,025		50·6	20·1	Fibrous
651	,, 104		1·11							
653	,, 106		1·11	0·968	49,330	50,961		51·8	21·6	,,
655	,, 108		1·11							
657	,, 110		1·11							
659	,, 112		1·11							
661	,, 114		1·11							
663	,, 116		1·11							
665	,, 118		1·11							
667	,, 120		1·11							
669	,, 122		1·11							
671	,, 124		1·11							
673	,, 126		1·11							
675	,, 128		1·11							
677	,, 130		1·11							
679	,, 132		1·11	0·968	50,240	51,901		48·0	20·6	,,
681	,, 134		1·11							
720	,, 200	Electric Welded.	1·11	0·968	50,245	51,906		50·6	23·1	,,
721	,, 201		1·11							
722	,, 202		1·11	0·968	49,090	50,713		46·8	20·8	,,
723	,, 203		1·12	0·985	53,310	54,122		52·7	25·6	,,
724	,, 204		1·11	0·968	50,055	51,710		49·4	25·1	,,
725	,, P		1·11							
			Total Mean of 32			52,484=23·4		50·5	22·2	

OF EXPERIMENTS.

Broke in Weld.					
Original.		Ultimate Stress.			Ratio of Weld to Solid.
Diameter.	Area.	Total.	Per Square Inch of Original Area.		
Inch.	Square inch.	Lbs.	Lbs.	Tons.	Per Cent.
1·12	0·985	41,765	42,401		80·8
1·11	0·968	45,340	46,839		89·2
1·11	0·968	39,245	40,542		77·2
1·11	0·968	48,860	50,475		96·2
1·13	1·003	47,420	47,278		90·1
1·11					
1·11	0·968	50,035	51,689		98·5
1·11					
1·11	0·968	49,860	51,508		98·1
1·10	0·950	46,635	49,089		93·5
1·11	0·968	44,835	46,317		88·2
1·11	0·968	47,340	48,905		93·2
1·09	0·933	39,915	42,781		81·5
1·12	0·985	47,205	47,924		91·3
1·10	0·950	43,880	46,189		88·0
1·11	0·968	38,810	40,093		76·4
1·11	0·968	48,240	49,835		95·0
1·12	0·985	47,255	47,974		91·4
1·12	0·985	48,510	49,249		93·8
1·11	0·968	45,090	46,581		88·8
1·11					
1·11	0·968	43,965	45,418		86·5
			Mean 46,899 =	20·9	89·3
1·14					
1·13	1·003	45,885	45,748		87·2
1·16					
1·16					
1·15					
1·11	0·968	43,720	45,165		86·1
			Mean 45,456 =	20·3	86·6

Results of Experiments to Ascertain the Behaviour under
Brand, Farnley. Electric Welds were

	Cold.					
Test No.	Description.				Angle.	Effects.
Y					Degs.	
404	Bar, 1⅛-inch diameter	F B	1	Electric Welded.	37	Broken.
406	,, ,,	,,	3		65	Cracked.
408	,, ,,	,,	5		65	Broken.
410	,, ,,	,,	7		34	Cracked.
412	,, ,,	B	9		58	Broken.
414	,, ,,	,,	11		115	,,
416	,, ,,	,,			65	Cracked.
418	,, ,,	,,	15		57	,,
420	,, ,,	,,	17		37	Broken.
422	,, ,,	,,	19		58	,,
424	,, ,,	,,	21		58	Cracked.
426	,, ,,	,,	23		50	,,
428	,, ,,	,,	25		90	,,
430	,, ,,	,,	27		150	Broken.
432	,, ,,	,,	29		95	,,
434	,, ,,	,,	31		59	Cracked.
436	,, ,,	,,	33		70	,,
438	,, ,,	,,	35		35	,,
440	,, ,,	,,	37		55	Broken.
442	,, ,,	,,	39		64	Cracked.
	Mean				66	
640	Bar, 1⅛-inch diameter	B	93	Hand Welded.	60	Cracked.
642	,, ,,	,,	95		180	Uncracked.
644	,, ,,	,,	97		90	Cracked.
646	,, ,,	,,	99		150	,,
648	,, ,,	,,	101		170	,,
650	,, ,,	,,	103		75	,,
652	,, ,,	,,	105		180	,,
654	,, ,,	,,	107		180	Uncracked.
656	,, ,,	,,	109		180	,,
658	,, ,,	,,	111		180	,,
660	,, ,,	,,	113		70	Cracked.
	Mean				138	
727	Channel, 1¾ by 1 by ¼-inch	B	82	Electric Welded.	28	Cracked.
728	Tee, 1¼ by 1¼ by $\frac{3}{16}$,,	,,	83		10	Broken.
729	Bar, tool steel, ⅝-in. square	,,	84		2 45	Broken, at weld. ,, in solid.
730	Angle, 1 by 1 by ¼-inch	,,	85		93	Cracked.
731	,, ,,	,,	86		35	,,
732	,, ,,	,,	87		70	Buckled, removed.
733	,, ,,	,,	88		75	,,
734	,, ,,	,,	89		75	,,
735	Bar, 1½-inch diameter, welded to tube ...	,,	90		8	Broken, cone weld.
736	Tube, 1½-inch diameter	,,	91		58	,,

726 Ring 13 inches diameter B 81. Bar cracked, clear

BENDING OF 72 PIECES OF IRON AND 1 OF STEEL, WELDED. BUTT; HAND-WELLS WERE SCARF.

	Hot.				
Test No.	Description.			Angle.	Effects.
Y				Degs.	
444	Bar, 1⅛-inch diameter	B 41	Electric Welded.	180	Very slightly cracked.
446	,, ,,	,, 43		180	Uncracked.
448	,, ,,	,, 45		160	Cracked.
450	,, ,,	,, 47		175	,,
452	,, ,,	,, 49		94	,,
454	,, ,,	,, 51		180	Uncracked.
456	,, ,,	,, 53		60	Cracked.
458	,, ,,	,, 55		180	Uncracked.
460	,, ,,	,, 57		96	Cracked.
462	,, ,,	,, 59		180	Uncracked.
464	,, ,,	,, 61		180	Slightly cracked.
466	,, ,,	,, 63		81	Cracked.
468	,, ,,	,, 65		163	,,
470	,, ,,	,, 67		98	,,
472	,, ,,	,, 69		180	,,
474	,, ,,	,, 71		90	,,
476	,, ,,	,, 73		120	,,
478	,, ,,	,, 75		180	Uncracked.
480	,, ,,	,, 77		180	Cracked.
482	,, ,,	,, 79		117	,,
	Mean			144	
662	Bar, 1⅛-inch diameter	B 115	Hand Welded.	100	Cracked.
664	,, ,,	,, 117		75	,,
666	,, ,,	,, 119		180	Slightly cracked.
668	,, ,,	,, 121		180	,,
670	,, ,,	,, 123		180	,,
672	,, ,,	,, 125		180	,,
674	,, ,,	,, 127		180	,,
676	,, ,,	,, 129		90	Cracked.
678	,, ,,	,, 131		180	Slightly cracked.
680	,, ,,	,, 133		180	Uncracked.
682	,, ,,	,, 135		95	Cracked.
	Mean			147	

of weld, when nearly straight. Bar 1-inch diameter.

PAPER II.

KOSHESHAH BASIN ESCAPE, MIDDLE EGYPT,

AND THE

BASINS BETWEEN ASSIOUT AND KOSHESHAH WHICH DISCHARGE THEIR CONTENTS THROUGH IT.

BY MAJOR R. H. BROWN, R.E.

1. *Subject.*—This paper deals with a definite section of the basin system of Upper Egypt, lying within the limits of what is sometimes called Middle Egypt, from Assiout, 228 miles south of Cairo, to Wastah, 51 miles south of Cairo. The length of the Nile valley included between these limits is therefore 177 miles.

In this length the Nile valley is, on an average, 12 miles broad, and in its breadth is divided up into (1) disconnected lengths of basin-land, east of the river, of insignificant areas; (2), the bed and islands of the Nile itself; (3), the tract under perennial (sefi) irrigation between the Nile and the basins; and (4), the chain of basins along the western desert and the course of the Bahr Yusuf (canal of Joseph).

It is with the last that this paper deals, but the tract under perennial irrigation affects the working of the basin system alongside of it, and must, therefore, itself, and its effects on the basins, be described so far as is necessary.

2. *Perennial Irrigation from Ibrahimiyah Canal.*—The perennial irrigation tract varies from two to six miles in width, and depends on the Ibrahimiyah canal for its irrigation. It is protected from

inundation on the side of the Nile by the Nile bank, and on the side of the basins by a longitudinal bank called the "Muhit." This tract begins at Mellawi, a town 82½ kilomètres (51½ miles) from the mouth of the Ibrahimiyah canal at Assiout; for it is at Mellawi, or a little to the north of it, that the water-level in summer-time first comes to the country surface. Above this point, on the south, as far as Derut, the left bank of the Derutiyah canal forms the eastern boundary of the basins; and from Derut to Assiout, the left bank of the Ibrahimiyah canal itself.

The canal Ibrahimiyah has an open head from the Nile to Assiout, and the flood waters enter it uncontrolled by any means of regulation until Derut is reached—61 kilomètres (38 miles) from Assiout.

At Derut are some fine regulators, by which the water is distributed, roughly, as follows:—

Maximum discharge of the Ibrahimiyah above Derut in high flood per 24 hours, 80 million cubic mètres.

DISTRIBUTION.

Heads opening directly on to the basins between Assiout and Derut... ...	10	million cubic mètres.
Dalgawi basin feeder	6	,, ,,
Bahr Yusuf	28	,, ,,
Canal Derutiyah	5	,, ,,
Canal Ibrahimiyah	10	,, ,,
Canal Saheliyah	3	,, ,,
Escape	18	,, ,,
Total	80	,, ,,

A plan of these works is given (*Plates* II. and III.), to show the general arrangement.

On the Ibrahimiyah canal below Derut there are regulators at

	Kilomètres from head.	Miles from Assiout.	
Minia	127	... 80	
Matai	169	... 105	
Maghaghah	197	... 123	
Feshn	221	... 138	new since 1884.
Beni Suef	254	... 159	

3. *State before Ibrahimiyah Canal was made.*—Before the Ibrahimiyah canal was completed, in 1873, for the irrigation of the former Khedive Ismail Pasha's extensive estates (now comprised in Daira Sanieh), on which sugar-cane was to be raised and sugar factories worked, the whole of Upper and Middle Egypt was under the basin system, the east boundaries of the basins and the Nile bank being one and the same. The basins were then fed direct from the Nile by the numerous canals, probably not much under control. It is most likely that in those days, when the Nile was high, there was a superabundance of water moving northwards through the basins, and breaches and cuts were numerous. But Upper Egypt suffers less from an excessive than from a defective Nile. Too much water does little harm ; too little water causes large areas to be left uncultivated for 12 months, and a loss of the revenue collected as land tax, since the tax is remitted from lands not reached by the water.

Being in direct communication with the Nile, and each basin probably having a special feeder of its own, the Nile water entered the basins freely, carrying its fertilizing matter in suspension with it, and leaving it in the basins, thus restoring annually to the soil what the last crop had taken out of it. After a good Nile these lands were then certain of producing a good crop.

The construction of the Ibrahimiyah canal parallel to the Nile and close to it, and the conversion of a strip of country into perennial irrigation between the Ibrahimiyah and the basins, separated the latter from direct communication with the Nile, and closed the special basin feeders.

Henceforward the basin chain from Assiout to Kosheshah had to rely on the system of basins north of Suhag to pass on water to it through the cross embankment at Assiout, and to the overspill of the Upper Ibrahimiyah and Bahr Yusuf, which latter takes off from the Ibrahimiyah at Derut. Thus the mouth of the Ibrahimiyah canal at Assiout was the most northerly point of supply for this chain of basins, extending 177 miles northwards from Assiout.

4. *The Bahr Yusuf and Adjacent Basins, and Basins Discharging into Bahr Yusuf. State after Construction of Ibrahimiyah and before 1884.*—Between Assiout and Kosheshah there are 23 distinct basins, besides the unembanked areas between provinces and on the west of the Bahr Yusuf. These basins vary in area from 8,000 to 50,000 feddans (feddan = 4,200 square mètres, nearly the same as an acre), with the exception of the last Kosheshah basin, which has an area of 80,000 feddans ; but this is really two basins made into one by the

suppression of the badly aligned embankment which used to separate the Bahabshin and Kosheshah basins ; the area of Bahabshin basin has thus been added to that of Kosheshah.

It is an advantage to have a basin of large area at the end of a chain of basins, and especially in the case of such an extensive chain as the one being considered ; for if any breaches occur at any point of the chain, causing a sudden increase of discharge into the terminal basin, the effect on this basin's water-level will be the less in increase of height, and the more gradual, in proportion to its size ; and the larger the basin the more time will be given for taking the necessary measures for escaping the excess water before it has time to raise the water-level in the basin to a dangerous height.

If, on the contrary, the terminal basin is of small area, a sudden flow into it, due to an accident, may cause it to rise rapidly, perhaps, during the night, and a breach in its banks might result, and all control of the water be lost, as breaches, such as sometimes occur in these large basins with their loose earth banks, cannot be closed, while the water continues to flow in large volume and at a high velocity.

The basins are enclosed by banks, generally five mètres wide at top, with two to one side slopes. The lateral boundaries are : on the east, the left bank of the Ibrahimiyah canal, from Assiout to Derut, of the Derutiyah from Derut to Kolobba Regulator, the Muhit from Kolobba to Ashment, and the railway from Ashment to Kosheshah escape ; on the west, the longitudinal banks along the Bahr Yusuf and the desert. Between these east and west boundaries the basins are separated from one another by cross embankments at intervals varying from 5 kilomètres to 16.

During the summer the crop of the preceding winter is collected at various spots alongside villages in the basins, and this must be removed safely out of reach of the water before the filling of the basin commences.

Besides these grain stores there are also considerable areas planted with durah sefi (millet), wherever water can be conveniently obtained either from hollows, wells, or other sources. The crop should be ripe about the 1st August, but in consequence of late sowing it is often not ready to be harvested before the 10th or 15th August. Consequently it is rarely possible to admit the water before the 10th August, which date may be taken as the present usual one for so doing.

Before the existing red-water feeders were made, these basins were filled as follows :—

On the 15th August, or as soon as the summer crop of millet and stores of winter-grown grain had been cleared away into safety, the Dalgawi head at Derut, and the three small heads between Derut and Assiout, and the head of the Bahr Yusuf at Derut, were fully opened. Later on the supply to the chain was supplemented by water being passed on through the bridges in the cross embankment of Assiout from the basins to the south of Assiout, which depended on another system of canals, mainly on the canal Suhagiyah, with its head of 21 openings of three mètres breadth at Suhag. This water, passed on from the southern basins, had deposited all its suspended silt before passing into the Kosheshah chain, and hence brought no fertilizing matter with it.

The supply of water from the above sources was kept up till about the 5th October, when, Assiout basins being full, and the time for discharging them having come, their water was passed off rapidly into the Bahr Yusuf, which was thereby raised to a sufficient level to flood the remaining parts of Minia and Beni Suef basins, if the discharge was effected to suit the requirements of the situation. The discharge was, however, carried out in a happy-go-lucky way, and often did not succeed in producing the result desired.

As far as the north limit of Assiout the basins, under the conditions described above, were not so badly off, but those of Minia and Beni Suef suffered considerably from their water having to come such a long journey after leaving the Nile. The Bahr Yusuf flows fairly red while the basins are filling, but its level does not rise high enough at first to flood more than the lower parts of the Minia and Beni-Suef basins, whose inundation had, therefore, to be completed by the discharge of the basins on the south. Thus it came about that the upper halves at least of all these basins were yearly inundated with water cleared of all its manuring properties, and year by year the evil effect of such a system was shown by the increasing inferiority of the crops raised.

The Bahr Yusuf water used to enter the basins alongside it by openings in the longitudinal banks at the south ends of the basins, and the levels in the basins were regulated to some extent by the scanty number of regulating bridges in the cross embankments. If these proved insufficient to dispose of the excess, and a basin rose above its safe limit, a cut into the Bahr Yusuf was made in the longitudinal embankment at a point sufficiently far removed to the south from the cross embankment to prevent too free a discharge, and lower the level too far before the full time for discharge had come.

The longitudinal section on *Plate* IV., showing the water levels of the basins when at full inundation level, and of the Bahr Yusuf at full supply, and also during the discharge of the basins, will show the reason of this.

From the section it will be seen that the ordinary full discharge of the Bahr Yusuf (full line) has a sufficiently high water surface to complete the inundation of basin Tahnashawi and all basins above it, and barely also that of basin Tahawi by admitting the water at its upper corner. To the north of basin Tahawi the Bahr Yusuf at full supply cannot directly fill any basin to full inundation level, but this can only be effected by passing the high-level water on from basin to basin, or still further raising the Bahr Yusuf level.

When the upper basins are discharged on to the Bahr Yusuf, the normal full discharge of the Bahr Yusuf is exceeded, and the water surface is raised to the level shown by the dotted line, which is high enough to complete the inundation of all the basins, though barely high enough for basin Garnusi.

It will be seen also that when the Bahr Yusuf is at its maximum level, due to the basin discharge, the Beni Suef basins cannot discharge into it, so that their discharge must be from basin to basin, through the bridges in the cross embankment, till the terminal Bahabshin-Kosheshah basin is reached.

5. *Remedies begun before* 1884.—The effect of the above-described faulty arrangements in causing deterioration of the soil had evidently attracted attention before 1884 (the first year of English control of the irrigation), for a syphon and canal (Nina canal), for direct supply from the Nile, had been built towards the north end of the chain, but had not been used for want of a slight effort to complete it. The stimulus was applied in time, and the canal was worked during the floods of 1884, the first beginning of an attempt to restore a red-water supply to the basins, in place of that which they had been unscrupulously deprived of.

There was another feeder also waiting completion. This branched off from the Ibrahimiyah canal, and had a masonry head built on the channel of the old Sabakhah canal, which used to feed basin Ashmunin from the Nile. On account of a fairly high rate of flow in the Ibrahimiyah canal, the next best thing to feeding the basins direct from the Nile was to feed them at different points from the Ibrahimiyah. This canal, however, is not large enough in its lower reaches to carry sufficient to feed more than one basin. But as far as the Sabakhah head its section is ample. Hence the Sabakhah canal project was a good one, but on account of obstruction on the part of land-owners, and the difficulty of getting proper work out of the corvée, it took quite two years before this canal was got into full working order.

Besides the above, no other project for giving red water to the basins seems to have been set on foot up to 1884.

6. *Remedies carried out since* 1884.—Since 1884 the following red-water feeders have been added to the two mentioned above :—

(1). From Derutiyah canal—Kolobba head—basin Ashmunin.
(2). From Nile—Abu Baqarah canal—basin Garnusi.
(3). From Nile—Sultani canal—basin Sultani.
(4). From Ibrahimiyah canal—Kolussi canal—basin Nuêrah.
(5). From Nile—Salim Pasha canal—basin Bahabshin.
(6). From Nile—Magnunah canal and two branches—Kosheshah.
(7). From Nile—opening in Kosheshah bank and Nasri bridge—basin Kosheshah.

There is also a project prepared, but not yet sanctioned, for a large canal from the Nile to enter Hod Deri, and another proposed for a canal from the Ibrahimiyah on Hod (basin) el Qurn.

Above Derut there existed previously three small heads on to the basins in the left bank of the Ibrahimiyah. These were not fully made use of, as they had imperfect means of regulation, and the engineers were at times unaccountably afraid to use them. These have been put in order, and the most made of them. To these three a fourth, and much larger one, has been added by adapting an old bridge, known as Mahgar-Mangabad, which lay conveniently close for the purpose in an old cross embankment.

The Nile feeder (3) in above list passes under the Ibrahimiyah by a newly-constructed syphon; No. (2) by an old syphon, which had to be dug out and restored; and No. (5) by an existing syphon, found in working order, though in bad repair.

No. (6) was brought into use merely by shifting the terminal dam of the Ibrahimiyah canal about half a mile to the south of the position in which it was found in 1884.

The manner in which No. (7) forced itself upon us as a feeder will be told further on.

7. *Discharge of Basin Feeders and Contents of Basins.*—The daily average discharges of each of the feeders to the Assiout-Kosheshah chain during the 50 days that the basins are filling (from 10th August to 29th September) are given approximately for a fair Nile

in the following table, which is a complete list of all the present sources of supply to the chain:—

Name of Feeder.	From what Source.	Discharge in million cubic mètres per 24 hours.
Bridge Gebel Assiout ...	Basin chain south of Assiout	2
Head Mahgar-Mangabad	Ibrahimiyah above Derut ...	4
Head Basin Ralbi ...	Do. ...	1½
Head Basin Rafi	Do. ...	1½
Head Basin Maharraq ...	Do. ...	1
Head Dalgawi	Ibrahimiyah at Derut ...	4
Bahr Yusuf	Do. ...	23
Head Kolobba	Derutiyah at Kolobba ...	3
Canal Sabakhah	Ibrahimiyah below Derut ..	4
Canal Abu Baqarah ...	Nile, by syphon under Ibrahimiyah	4
Canal Sultani	Do. ...	2
Canal Nina	Do. ...	1
Canal Kolussi	Ibrahimiyah below Maghaghah	1
Canal Salim Pasha ...	Nile, by syphon under Ibrahimiyah	2
Canal Magnunah	Nile, by direct canal	2
Kosheshah Escape ...	Nile direct	Nil.
	Total ...	56

The amount that enters through the Kosheshah escape is taken as "Nil," as the amount entering during the early part of the 50 days is probably counterbalanced by the amount discharged during the latter part through the Kosheshah escape, and through a 7-arch bridge in the cross embankment of Kosheshah basin.

The Fayum draws off from the Bahr Yusuf six million cubic mètres a day. Deducting this, there is discharged into and remains in the basins 50 millions × 50 days = 2,500 million cubic mètres.

This calculation agrees with that made by Colonel Western when he was making his calculations for the design of Kosheshah escape. He arrived at the quantity of water to be discharged as follows :—

"The series of basins to be emptied by the escape comprise a total area of 555,652 feddans (acres nearly), from Assiout to Kosheshah, and the total maximum volume required to flood these lands may be taken as 2,445 million cubic mètres, equivalent to an average rate of 4,400 cubic mètres per feddan (4,200 square mètres), or an average depth of 1·05 mètres. But of this 2,445 millions some 200 are set down as being expended by evaporation, left in hollows, etc., and 245 millions as discharged through the 7-arched bridge into basin Riqqah (next basin on the north to basin Kosheshah), and through Lahun bridge into the Fayum.

"(1). We have then for the Abu Khadigah (Kosheshah) escape a maximum discharge of 2,000 millions, and the time allowed has been fixed at 20 days.

"(2). The maximum volume for low years is estimated at 1,500 millions, and the time given for discharge 10 days.

"The *average* discharge, then, required of the Abu Khadigah escape are 100 million cubic mètres per day in high flood years, and 150 millions in low floods."*

The reason for this difference between high and low Niles is that to obtain favourable conditions for the crops to be raised in the basins, the water must not be run off *before* a certain date, or remain covering the ground *after* a certain date. From these considerations it has come about that the usual date for cutting the Kosheshah bank to let the basin waters off has been from the 15th to 20th October. The earliest date recorded is the 7th October, in 1887, a year of high Nile; the latest, the 29th October, in 1888, a year of low Nile, excepting 1878, an abnormal year of high and prolonged flood, when it is recorded that the cut was made on the 7th November, and it is supposed not to have been made earlier, as the Nile remained so high that there was not sufficient difference of level between the basin and the river to make the cut before that date. The crops suffered in consequence.

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* Extracted from Colonel Western's "Report to accompanying design and estimate for an Escape Dam at Abu Khadigah to Abu Kosheshah basin series."

Further, in a high Nile year all the basins are full early, and the discharge from Kosheshah can begin early. In low Nile years the basins are not full at the time for discharge, and the inundation of one basin after another is completed from the next basin above it; hence the discharge from Kosheshah cannot begin till late. But in both cases the ground should be clear of water by the same time. Hence, in a high Nile year it has been calculated that 2,000 million cubic mètres must be discharged in 20 days, and in a low Nile year 1,500 millions in 10 days.

8. *Description of Method of Discharging Basins.*—But before going further into the calculations on which Colonel Western based his design of the Kosheshah escape, it will be as well to give a short description of the system of discharging the basins as practised by the Egyptian engineers, and the modifications introduced by the English control of them.

As described before, the basins of Assiout become full during August and September, and towards the end of September it was the custom for the chief engineers of the different provinces in which the basins lay, to meet together with mudirs (governors) and omdehs (chiefs of villages), and probably a delegate from the Minister of the Interior, to draw up, after much talk, a programme for discharging the basins. This description relates to the period previous to the appearance in 1884 of English inspectors of irrigation in the districts. Since 1884 the inspector has taken the place of the mudirs, omdehs, and delegate of the interior, and much of the talk, and instead of the chief engineers putting their seals to the programme drawn up, to show that they consent to be bound by it, they receive their orders from the inspector to carry out the programme which, after discussion with them, he judges to be the best.

The programme used to lay down that the water should be handed over from one province to another on such and such dates, and on these dates the two chief engineers concerned, with a retinue of assistants and district engineers, used to meet on the dividing bank between the two mudiriyahs, the mudir or his representative being present, and sometimes also a delegate from the Minister of the Interior, each, of course, with his attendant retinue. A cut in the bank was then made. Then, by dictation, subjected to much correction by the different officials present, a record of the proceedings got drawn up by a liberal allowance of clerks, to which all the officials attached their seals, and the chief engineer of the north signed a receipt for the waters handed over to him, and gave it to the chief engineer of the south. And then all the engineers went home, and did nothing till the next Nile flood. At least, such is the account they give of themselves when they are contrasting the hard work they have to do now all the year round with what they used to do.

The necessity for these elaborate ceremonies ceased with the introduction of more discipline. The chief engineers now receive their instructions from the inspector as to the way in which they are to discharge the basin, and they are expected to carry them out.

The discharge used to be effected by freely cutting all the cross embankments of the basins, one after the other, often at several points, whether the existing bridges were of sufficient waterway to do without cuts or not. There seemed to be an unsatiable appetite for cuts. There was no calculation and no system in this operation. The principles of basin irrigation, as they have been gradually developed in Upper Egypt (where not interfered with by the Ibrahimiyah canal), may be worthy of admiration, but skill and order in conducting the operation of filling and discharging them (though lately credited to the former Egyptian engineers in a book on Egyptian irrigation) did not exist.

Instead of cutting the lateral bank and discharging into the Bahr Yusuf, and so getting rid of the water once for all, the cross embankments were freely cut, and the whole body of water passed through the basins from end to end of the chain, necessitating an increased number of cuts in each cross embankment as the water travelled from south to north.

Most of these cuts have now disappeared. By making use of the Bahr Yusuf as the main channel of discharge, a cut in a cross embankment is now rarely made, and only for a special reason; but for want of sufficient masonry escapes, cuts have still to be made in the west lateral banks of the Minia basins to discharge them on to the Bahr Yusuf.

9. *The Kosheshah Dam, and the Effect of Cutting it.*—By the discharge of the basins north of Assiout, all their water at last finds its way into Kosheshah basin, which, about the middle of October, commences to rise more rapidly, till at last it becomes necessary to open the cut of discharge in the Kosheshah bank on to the Nile about the 20th October.

This is the cut, which has now been replaced by a masonry escape, the subject of this paper. By the sudden release of the enormous body of water retained by the dam, the Nile is raised sometimes sufficiently to flood the high part of Lower Egypt, which the unassisted true flood level had failed to reach. The rise of the river due to this discharge varies with the level to which the river has fallen, and with the extent to which the basins have been filled.

In 1884, a year of low Nile in October, the cutting of the Kosheshah dam caused the river at the barrage below Cairo to rise 1·42 mètres in 30 hours, representing an extra discharge of 210 million cubic mètres a day, according to Mr. Willcocks. This artificially-created flood is considered by Lower Egypt of considerable value, and in designing the Kosheshah escape the production of this rise had to be provided for.

The diagrams of the barrage gauge readings for the periods that the Nile level was affected by the Kosheshah discharge, given on *Plate* V., are interesting. Those for 1888 and 1889 were made out by Colonel Western from hourly readings, with the object of checking the quantities that the Kosheshah cut was supposed to discharge. The cut would, of course, be discharging more than the figures calculated from these diagrams give while the level was rising, and less while the level was falling.

The diagrams for other years are made out from the readings on the barrage gauge, taken daily at 6 a.m.

In 1884 and 1888, as large a wave as could be produced was desired by Lower Egypt; in 1887 and 1890, Lower Egypt wished for no further increase to the Nile level in October. The discharge of the basins and cutting of the Kosheshah dam were, therefore, arranged in such a way as to secure the result desired as nearly as possible, and, as the diagrams show, with success.

10. *The Old Kosheshah Dam, alias Abu Khadigah and Nasri.*—The cutting of the Kosheshah dam was always made an occasion for what is called in Egypt a "fantasia," and in India a "tamâsha," and was looked upon as the final operation of the Upper Egypt basin discharge. A breach of this bank before the proper time was considered too awful a thing to contemplate, and the result of such a breach could, it was thought, be nothing else but a widespread calamity, from extensive areas of land being deprived of the power of raising their annual crop for want of being inundated.

There appears to be no record of breaches previous to 1884, but it would be strange if the bank had never breached, seeing that a new bank has to be made yearly to close the breach of the former year, the height of this bank varying from 7 to 10 mètres, and the base of it being sometimes formed in water. After 1884, its section was made six mètres wide at top, with two to one side slopes; before 1884 the crest was probably wider, but the slopes steeper.

It was only to be expected that such a bank should break some time or other. If it had breached previous to 1884, no doubt large numbers of corvée would have been called out, and the breach, if possible, closed. But there are no records as to whether it ever breached.

In 1884, the dam stood firm, and in consequence of an absence of instructions, it was not cut till Kosheshah basin had risen considerably above its ordinary full flood level. The Nile was rather a low one, so that at the time the dam was cut there was a head of 2·50 mètres on it. The rise at Cairo due to this cutting has been referred to before.

During the summer of 1885 this breach was closed as usual. On the 2nd August news reached the irrigation headquarters that the river had burst through the dam into the basin, which at that time had not begun to fill from the Bahr Yusuf, its then only source of supply. This was startling news for the English irrigation officer, who had lately joined the department, and was in temporary charge while the inspector of irrigation was on leave. He very wisely did not at once proceed to close the breach, but to calculate the consequences of leaving it open, lining the bottom and sides of the opening with dry rubble pitched in to check its further widening. A considerable amount of stone was collected ready to close or partly close the opening if the respective rise and fall of the basin and river rendered it desirable. But it was not necessary to use the stone, as the basin successfully rose to the required level in spite of the continually increasing discharge into the Nile through the open breach. As soon as the basin had reached its full inundation level (and by good luck not before), the sides of the cut gave way, and the breach rapidly widened.

During all the time that the basin was filling, the decision to leave the cut open was denounced by native governors, engineers, and notables, as convincing proof of lunacy on the part of the English engineers; "bêtise" was the French expression used to describe it.

The beneficial effect, however, of the direct entry of the Nile water through the breach to lands deprived of red water for perhaps 15 years, was so marked in the quality of the crops raised, that in 1886 the land-owners petitioned Government to leave the breach open and repeat the act which they had denounced the previous year as folly.

It had been decided to leave an opening for the entry of the Nile water before the petition was made. The widened opening of the previous year was narrowed to 40 mètres base, and given a floor built in dry rubble, the two cheeks of the opening also being revetted at one to one slopes with dry rubble. Stone was collected also on each side to form a closure with it if necessary. This arrangement succeeded in its object without a closure being formed, the Nile levels being favourable, but for a few days before the basin rose to its full level, a day-and-night struggle was carried on with stone and sacks filled with earth, to prevent the opening from widening out too soon.

The continuous flow through the opening dug out a hole 27 mètres below water surface, that is, to one mètre below mean sea level, the country level at Kosheshah being about R.L. 24·00.

Again, in 1887, the same arrangements were made as in 1886. The year 1887 was a year of very high Nile, and the problem was how to let off the basins without causing a rise of the river at Cairo. Consequently, as soon as the river fell below the basin level, a cut in the bank was made in addition to the opening, and the basin allowed to fall gradually with the river.

In consequence of the difficulty experienced in 1886 in keeping the opening from widening too soon, and recognizing that it was good fortune that prevented it from doing so in 1885, and that the Nile levels might not always favour us, it was decided, in 1888, to form a platform higher than in previous years, with its top surface low enough to allow the entrance of the flood water, but as soon as the river and basin should become one level, to close across this opening with stone, to avoid the danger of the opening being widened out by the return flow into the river before the basin had risen sufficiently to flood its highest parts. This plan was carried out, and fortunately too, as the Nile of 1888 was a very low one, so low that no waste of water from the basins could be allowed, and the stone bank, after being formed as intended, had to be made tight by adding a covering of grain sacks, filled with earth, in front of it.

The head against this dam was 4·40 mètres, when the bank was cut to discharge the basin.

By 1889 it had been decided to build a masonry escape, and it was thought wisest to close the cut with earth, as usual, but not to re-open the stone dam, and to be satisfied with allowing water to enter from the Nile by an old 4-arch bridge in the Kosheshah bank.

The same was done in 1890, during the summer of which year the floor of the masonry escape was built. The earth dam was cut, as usual, in 1890 for the last time. The old Kosheshah bank, the victim of so many cuts, has now been cleared away to form the banks at each end of the Kosheshah escape.

11. *Colonel Western's Calculations* (*continued*).—In paragraph 7 it was shown how the conclusion was arrived at that the masonry escape to replace Kosheshah cut must, in a high Nile year, discharge 2,000 million cubic mètres in 20 days, and in a low Nile year, 1,500 million cubic mètres in 10 days.

Colonel Western's report continues the calculations as follows:—

"From a study of the dates of cutting Kosheshah bank in former years we may assume, for purposes of calculation, that the basin escape will be opened on October 22nd, or 19 days before November 10th, the date laid down for the completion of the discharge.

"From a comparison of levels it is found that the heads, at the time of opening, will vary from 0·30 to 4·50 mètres, and with these heads, less the rise of the river consequent on the discharge, the escape must be designed to pass 100 and 150 million cubic mètres per day.

"It may be remarked that in order to give an *average* discharge per day of 100 and 150 millions, the first discharge must be increased by one-half, or to 150 and 225 millions; but, considering that the basins have never to date been all full at the same time, and as the cost of the work has been limited to £60,000 or thereabouts, it will be sufficient to calculate for the average discharge for the first outflow, and then arrange, as far as possible, for the maintenance of these same discharges.

"Minimum spring level at site of work may be taken as at R.L. 19·00, and foundation line must be below this, or, say, at R.L. 18·50. Floor line may then be placed at 20·50 or above. Assuming this level of 20·50, basin level at 26·70, and river at its minimum 22·20, it will be manifestly injudicious to allow, on first opening, the full depth for discharge of 26·70 – 20·50 = 6·20 mètres into a backwater of only 22·20 – 20·50 = 1·70 mètres + rise of river, say, 1·00 = 2·70 mètres.

"The depth of opening must then be divided into two; the first, or upper series, to be opened to discharge the volumes demanded, 100 and 150 millions; and the lower series to be kept as a reserve, and opened only to maintain the discharge as the head decreases, or water surfaces in basin and river fall.

"These lower sluices will also be available for filling the Kosheshah basin during the rise of the river, the volume to be passed in being estimated at 400 million cubic mètres.

"From various trial calculations the upper sluice-gate has been fixed at three mètres in depth, or from R.L. 26·70 to R.L. 23·70. Assuming this depth, and the formula for discharge in cubic mètres per second, discharge

$$= \tfrac{2}{3}\, l \times 4{\cdot}43 \sqrt{h}\, (d + \tfrac{2}{3}\, h),$$

or, for a length of one mètre

$$= 2{\cdot}953 \sqrt{h}\, (d + \tfrac{2}{3}\, h),$$

we shall have discharges as follow, with basin gauge at 26·70 :—

River at	Million c.m. per day.	M³. per second.	Mètres of opening required for 100 Mn. c.m. discharge.
26·40	0·4054	4·692	246
26·00	0·5920	6·846	169
25·70	0·6814	7·887	147
25·40	0·7480	8·655	...
25·00	0·8089	9·323	...
24·70	0·8419	9·745	...
			Mètres of opening required for 150 Mn. c.m.
24·40	0·8643	10·004	174
24·00	0·8806	10·192	170
23·70	0·8841	10·233	169

and below.

"Now the river has only been recorded as above R.L. 26·0 on two occasions during the last 30 years, whilst a large discharge with a low river is most important. It is evident, then, that the 174 running mètres of waterway for 150 millions should rule the length.

"Assuming a round number of 180, 36 bays of five mètres, or 60 bays of three mètres, are required. (It was decided to build it of 60 bays of three mètres). With the basin at 25·70, we have the following figures :—

Raised level before discharge.	First discharge, million c.m. per day.	Raised river level in consequence of discharge.	Decreased discharge, million c.m. per day.	Remarks.
26·00	106½	26·40	73	Must be supplemented by partially opening lower series.
25·50	131	26·00	112	
25·00	145½	25·70	123	
24·00	158½	24·90	148	
23·00	159	24·10	158	
22·50	159	23·70	159	

"The effect on the level at the barrage of opening at the above levels is shown in the following table :—

Basin gauge.	River gauge at Wastah, five kilomètres below Kosheshah.		Corresponding to Barrage.		Rise at Barrage.
	On opening.	After opening.	Before opening.	After opening.	
26·70	26·00	26·40	17·50	17·90	0·40
,,	25·50	26·00	17·00	17·50	0·50
,,	25·00	25·70	16·50	17·20	0·70
,,	24·00	24·90	15·50	16·40	0·90
,,	23·00	24·10	14·50	15·60	1·10
,,	22·50	23·70	14·00	15·20	1.20

"The design for gates has been selected with a view to simplicity, combined with quick opening for the upper tier, and preference has accordingly been given to horizontally pivotted falling gates for upper series, and direct lifted gates in vertical grooves for the lower sluices."

(End of extracts from Colonel Western's report).

12. *Manner of Working Escape.*—The escape will be worked as follows (see *Plate* VI.):—

Before the river rises, the upper drop gates will be raised to close the upper openings and secured by their hooks and eyebolts, built in the masonry. The lower gates will be raised in the vertical grooves till they are clear of their openings, and will be fixed in that position.

As the Nile rises, it will enter by the under-sluices and help to fill Kosheshah basin. As soon as the basin and river come to one level, and the return flow out into the river begins (without a probability of the flow being again reversed by a further rise in the river), the lower sluice-gates will be lowered and the lower sluices closed (a bottom sluice-gate weighs 1·75 tons, and will close by its own weight under a pressure due to a head of water of 1·30 mètres). The basin will then rise further, and if the time for discharge has not come, and it is necessary to let off some of the basin water, the lower sluices can be partially opened to regulate the level of the basin; but each sluice will not be raised beyond that amount that will bring its upper edge on a level with the top of the sill which lines the down-stream edge of the upper floor, so that the lower gates may not interfere with the discharge through the upper gates when they are let go.

When the time for the general discharge has come, all the upper gates will be let go, and a day or two more afterwards the lower sluices will be fully opened to assist the discharge.

To prepare the upper gates for letting them go, the strain is brought on to two chains, which are secured by a clip link, and the hooks which took the strain hitherto are lifted out of their eyes. The gates are then released by knocking off the clip link (see *Plate* VII.).

The weight of each upper gate is 2·70 tons.

13. *Modification of above Programme.*—A modification of this programme will probably be adopted in years when, at the time for letting go the upper gates, the Nile level has fallen below the level of the upper floors. In such a case the lower gates will be raised till their upper edges are on a level with the top of the sill which lines the down-stream edge of the upper floor. The flow through the under-sluices will cause the down-stream (river) level to rise, and the up-stream (basin) level to fall, and as soon as it is considered that the head has been sufficiently reduced to make it safe to do so, the upper falling gates can be let go.

Plate VIII. shows the design of the Kosheshah escape.

ADDENDUM.

Opening of Escape in 1891.—Since the foregoing was written, the Kosheshah escape has been worked through its first flood season with success. The ceremony of letting go the gates for the first time took place on the 17th October, 1891, the Minister of Public Works representing His Highness the Khedive on the occasion.

Previously to the 17th October, 32 of the bottom gates had been raised 75 centimètres gradually, to keep the basin level down to that of full irrigation. This reduced the head of water on the gates from 1·14 mètres to ·80 by raising the level of the river below the escape. The discharge of some of the largest basins south of Assiout, and the closing of the heads of some basin canals, had been so timed that their maximum effect in raising the river was calculated to be felt at Cairo on the 18th October, and to coincide with the increase due to the opening of Kosheshah escape. It is desirable in low Nile years to produce by means of the discharge of the basins a level of 23 dhras on the Rodah gauge (a little above Cairo), corresponding to a level of 17·18 on the Rosetta barrage. There was produced in 1891 22 dhras 20 kirats on the Rodah gauge, and 17·09 on the barrage gauge, being a level short of that which is considered the most favourable by about four inches only.

On the 17th October the mass of the basin waters which had been discharged into the Bahr Yusuf from the Assiout and Minia basins had not yet reached Kosheshah, and did not arrive till about the 21st. Had the opening of the escape been fixed for the 21st, perhaps 23 dhras at Rodah would have been reached, if not exceeded.

The necessity, however, for having a fixed day announced some time beforehand for the ceremony of inauguration, prevented the postponement of the discharge till the date which, later on, appeared more advantageous.

Plate V., which shows the effect at the barrage for different years, shows that obtained in 1891 also.

On the 17th October, 1891, all of the 60 upper gates were released, within a period of 20 minutes, by four parties of three each, without a hitch. The whole number could have been let go in less than 10 minutes, but an interval of 10 minutes was given to allow the spectators to alter their positions. Probably two men alone could have let go all the 60 gates under half an hour.

As soon as the upper gates were released, those lower gates, which had not already been raised 75 centimètres for purposes of regulation, were raised to that extent, the raising being completed by midday on the 18th. As the rise at first was insufficient to satisfy Lower Egypt, orders were given on the afternoon of the 18th October to raise all the lower gates as high as possible. This was completely carried out by the 19th October. Though by so doing some of the waterway of the upper openings was obstructed, yet more waterway was gained than lost.

It so happened that from the 15th to the 19th October a rise of the river of seven kirats (nearly 16 centimètres) took place at Aswân. The effect of the Upper Egypt basin discharges added to this, and the closing of the feeder canals caused a rise at Minia from the 15th to the 22nd October of 53 centimètres, and maintained the level reached on the 22nd for three days, till the 25th, when the final fall commenced.

The Wastah Nile gauge, about three miles below Kosheshah escape, rose 80 centimètres from the 15th to the 24th October, and the Rodah (Cairo) gauge rose 90 centimètres from the 11th to the 24th October, the level reached being maintained on the 25th October, after which the final fall commenced.

The upper drop gates were a novelty in Egypt, and this fact, combined with the size and importance of the escape, attracted a considerable number of spectators, a special train from Cairo bringing a distinguished load of Ministers and high officials, among whom were the following Royal Engineers, specially mentioned here as this paper is written expressly for the R.E. Institute :—

Colonel Sir Colin Scott-Moncrieff, K.C.M.G., C.S.I., as Under-Secretary of State for Public Works.

Colonel J. C. Ross, C.M.G., as Inspector-General of Irrigation.

Colonel H. H. Kitchener, C.B., C.M.G., Egyptian Army.

Lieut.-Colonel H. H. Settle, D.S.O., Egyptian Army.

Captain W. F. H. Stafford, Army of Occupation.

The writer of this paper was also present as Inspector of Irrigation in charge of the work.

APPENDIX.

Cost of Kosheshah Escape.

Description of Work.	Quantity (cubic metres).	Cost (Egyptian pounds).
	c. m.	L. E.
Excavation in foundations ...	68,708	2,102·491
Earthwork in approach channels and banks	154,300	4,976·812
Brick masonry	7,688	7,719·624
Rubble masonry, ordinary ...	17,768	14,214·700
Rubble masonry, in Portland cement	2,108	3,373·654
Concrete	146	146·250
Ashlar masonry	1,324	5,694·050
Dry rubble masonry	16,245	5,264·686
Unwatering	...	1,152·000
	Lineal mètres.	
Piling	549	3,322·530
Ironwork	Quantities, next page	13,413·372
Woodwork...	...	560·327
Hutting	..	573·694
Sundries	...	105·410
	Total	L. E. 62,619·600

There was also charged against the work the cost of temporary arrangements for controlling the water, not fairly chargeable against it, and, therefore, omitted above. This expenditure was—

	L. E.
Construction of dry rubble-stone dam ...	3,267·300
Removal of ditto	150·231
	3,417·531
Less value of Government materials supplied	414·750
Debited to the escape accounts	3,002·781

Rates paid for work.—The rates paid for different descriptions of contract work were the following :—

Earthwork...	$3\frac{10}{40}$	piastres	a cubic mètre.
Brick masonry	100	„	„
Rubble masonry, ordinary	80	„	„
Do. in cement	160	„	„
Dry rubble masonry ...	55	„	„
Do. with Government stone	15	„	„
Ashlar masonry	430	„	„
Piling	600	„	per lineal mètre.
Wooden gangways . .	700	„	per arch.
Timber for winch railway	20	„	per cubic foot.

Ironwork.

	Quantities.	Rates. P.E.	Cost. L.E.
Wrought-iron ...	367·50 tons ...	1,975 ...	7,258·000
Cast-iron	205·80 „ ...	1,425 ...	2,932·000
Timber, teak ...	381·60 cub. ft....	100 ...	381·600
Iron rails	573·30 yards ...	20 ...	114·600
Phosphor bronze ...	6,360 lbs. ...	9 ...	572·400
Chains	11·057 tons ...	2,050 ...	226·608
Steel	4· 58 „ ...	2,470 ...	113·126
Felt...	1,172 sq. ft. ...	... ...	10·000
		Total ...	11,608·334
Winches	2 No.	L.E. 668·067 ...	1,336·134
			12,944·468
Other iron and cost of erection		... ...	468·904
		Total cost, ironwork ...	13,413·372

PLATE I.

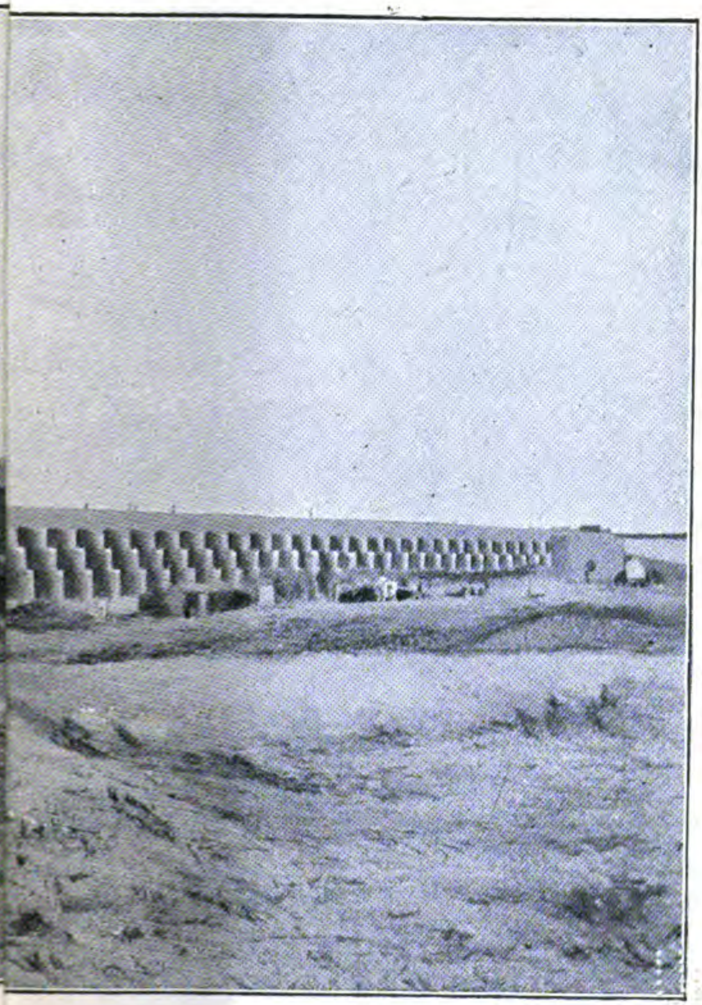
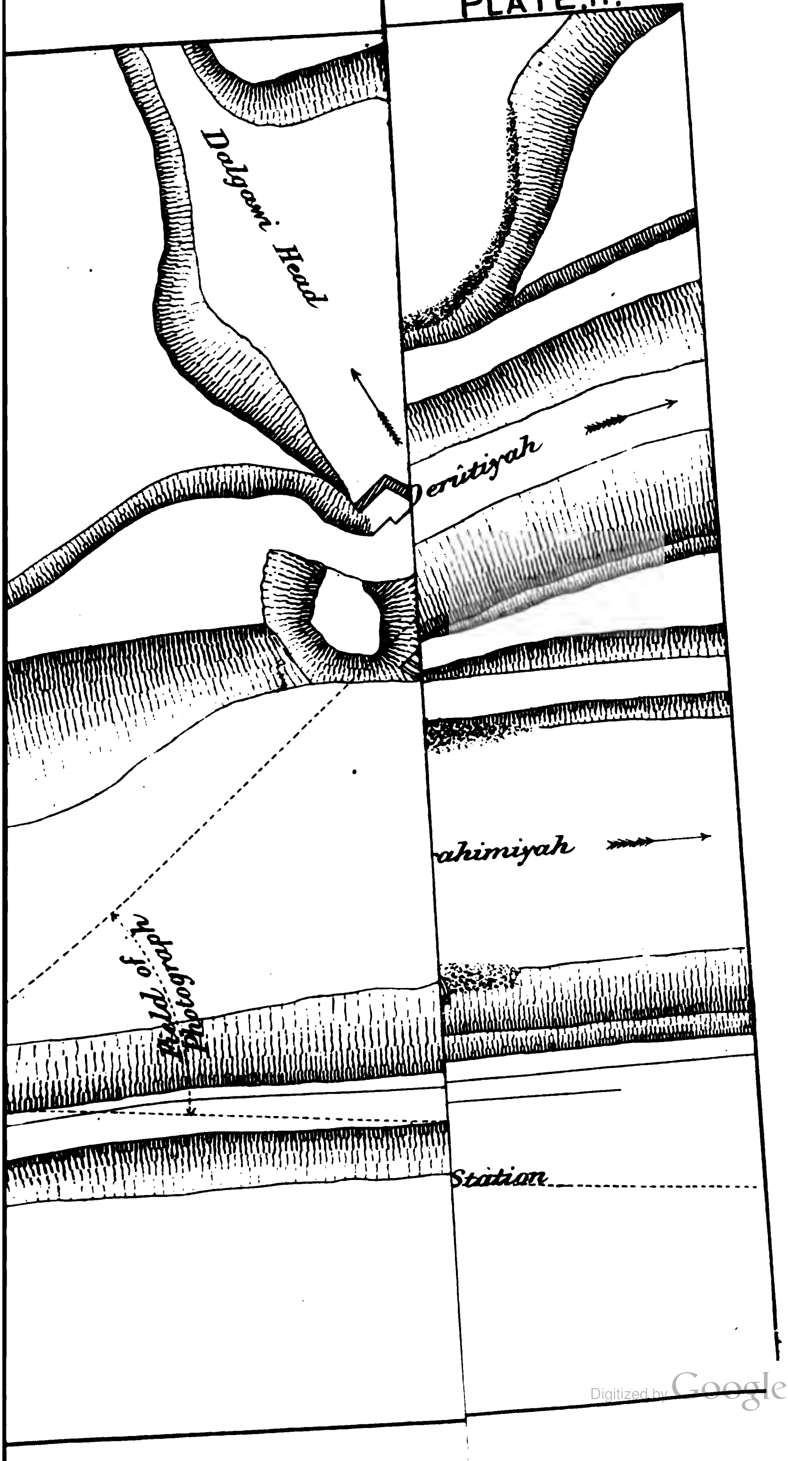


PLATE II.


Dalgomi Head
Derûtiyah
ahimiyah
Field of Photograph
Station

EGULATORS. PLATE III.

PLATE. IV.

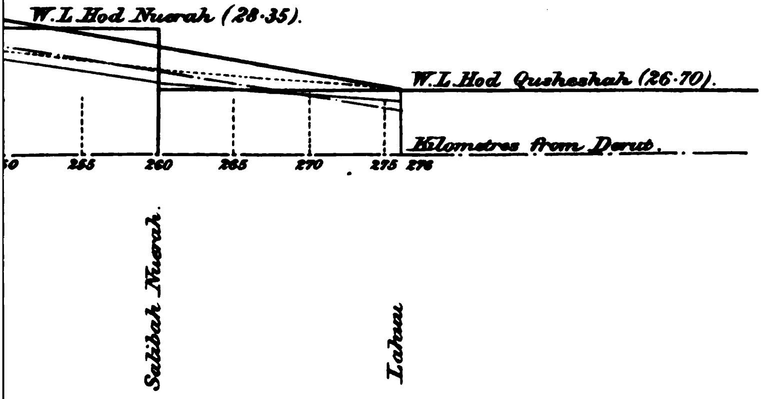
n discharge, on 30th Sept 1891.. Bahr Yusuf level at Derut 46·60.

Asyût & Minia Basins, on 16th October, 1891.

syût & Minia Basins, on 20th October, 1891. when water in

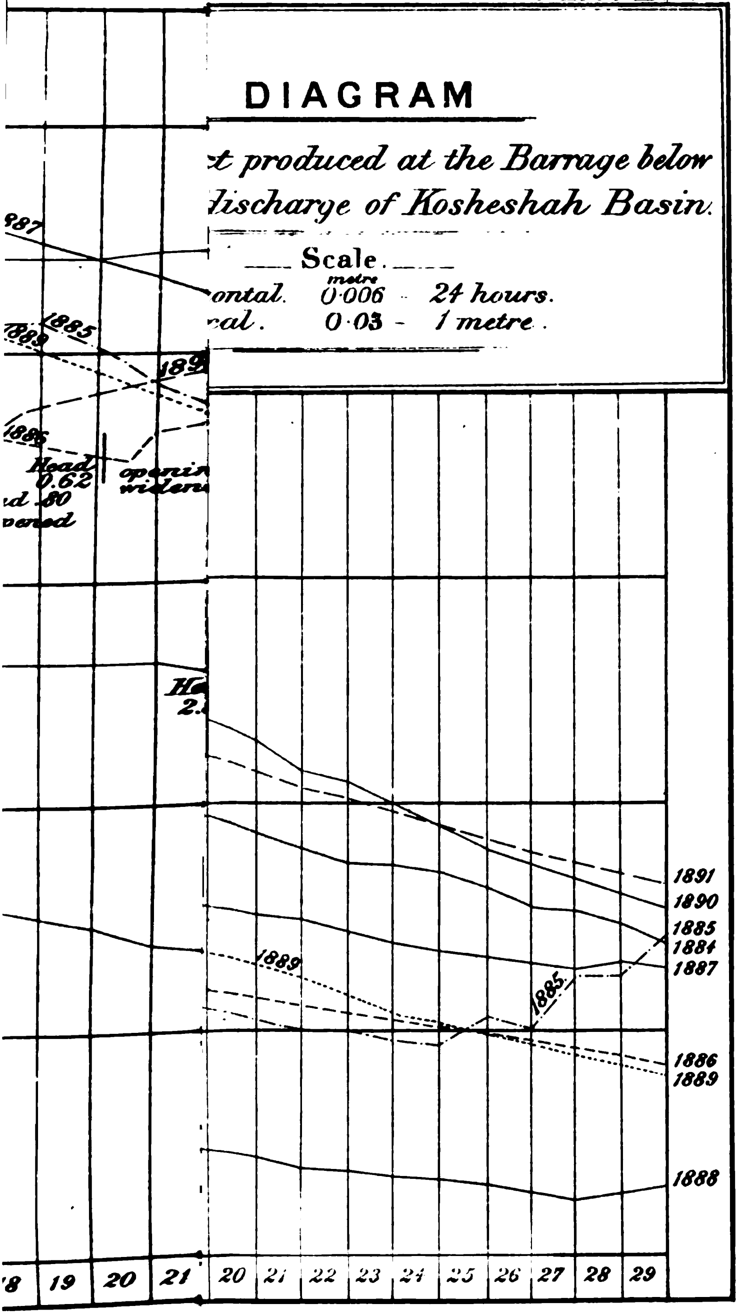
t points in each Basin (approximate).

).



ASINS.

PLATE.V.



DIAGRAM

ct produced at the Barrage below
Discharge of Kosheshah Basin.

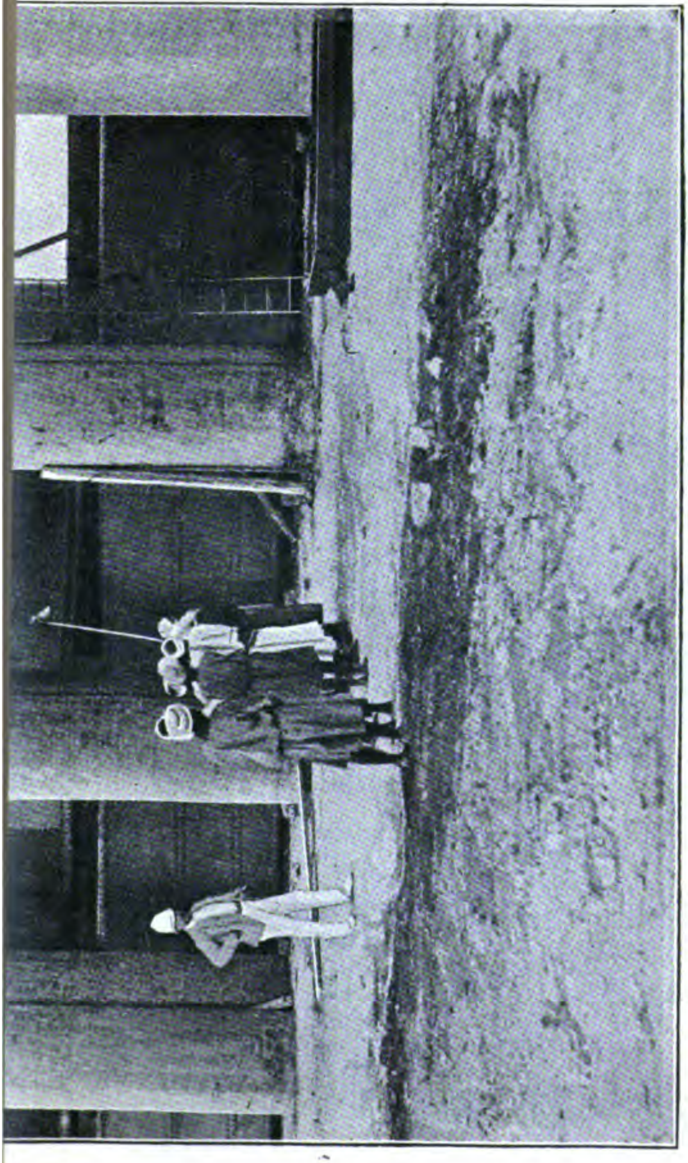
Scale.

ontal. 0·006 metre - 24 hours.
cal. 0·03 - 1 metre.

PLATE VI.

ROM BASIN SIDE,

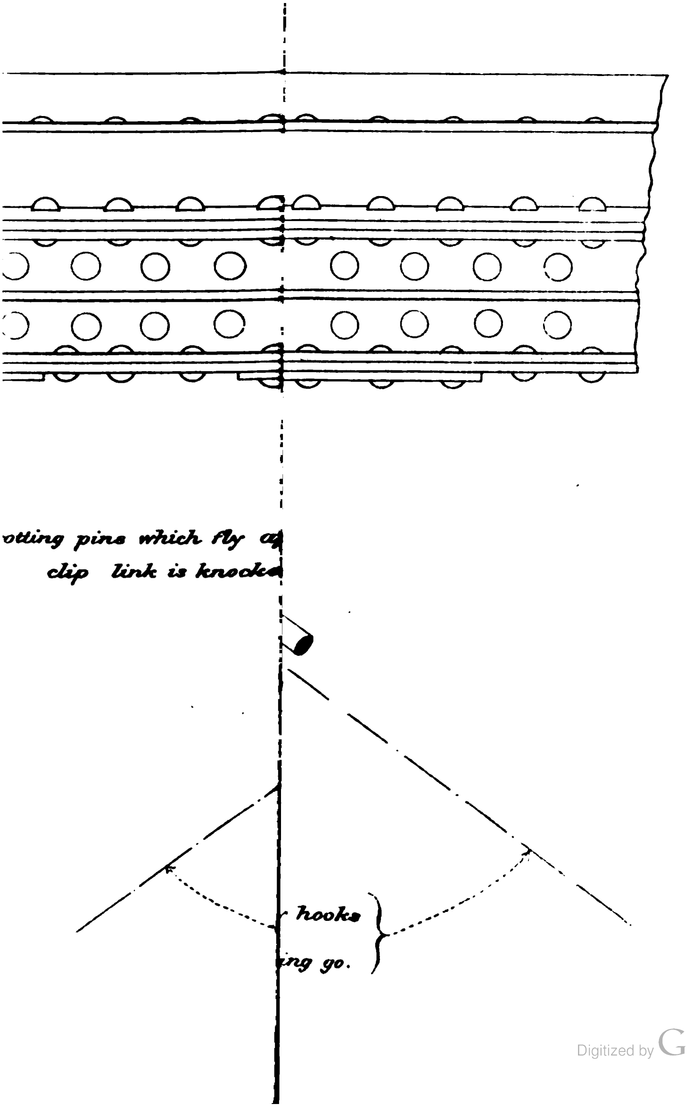
during construction, the Lower Gates being
r Grooves.



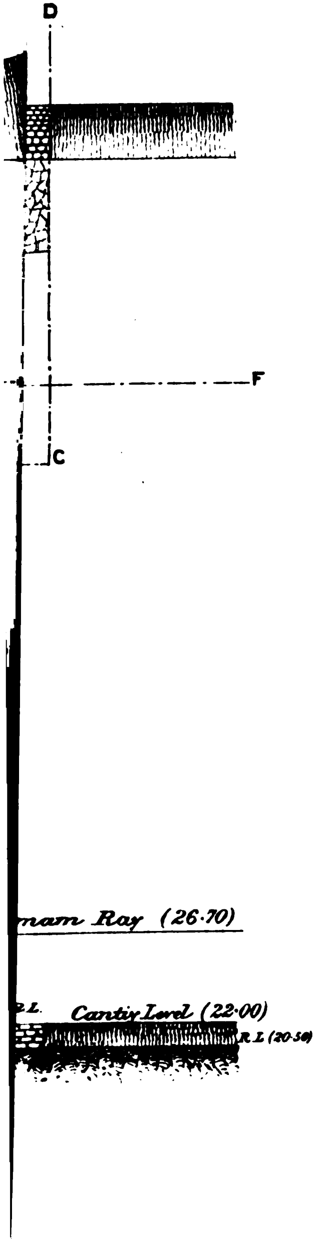
ISED. SHOWIN

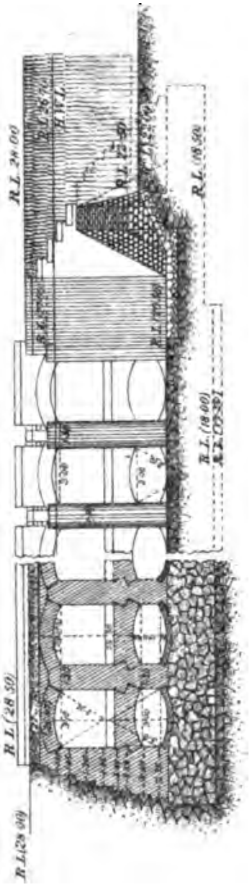
PLATE. VII.

hes = 1 Foot.



PLATE, VIII.





A·B C·D.

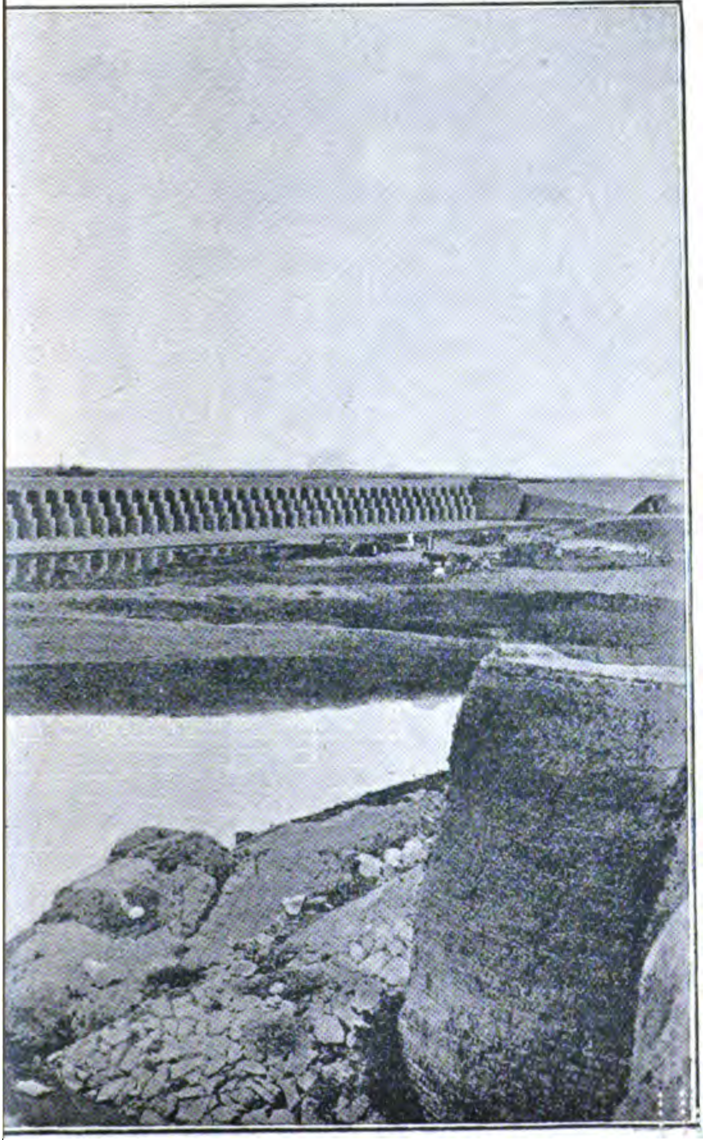
HES

k betw

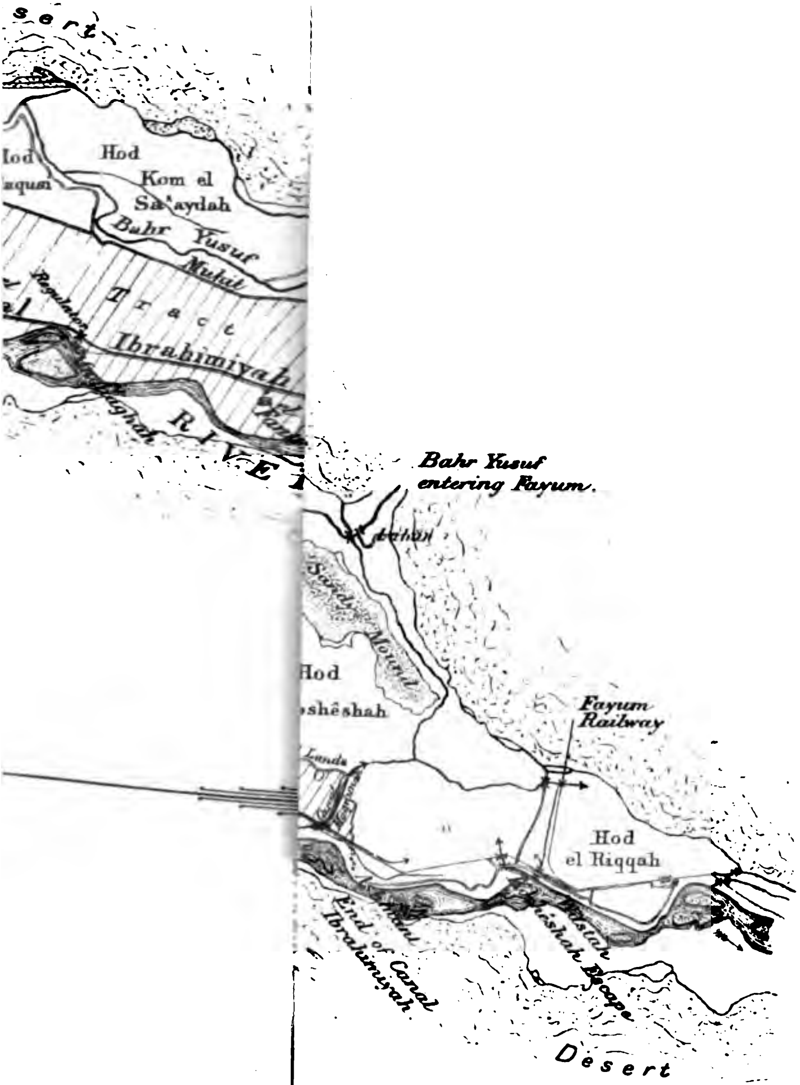
PLATE IX.

HESHAH ESCAPE

k between the New Escape and the River.



PLATE, X.


sert
Hod
Kom el
Sa'aydah
Bahr Yusuf
Muhit
Tract
Ibrahimiyah
Regulator
Bahr Yusuf
entering Fayum.
Sands Mound
Hod
Lands
Fayum
Railway
Hod
el Riqqah
End of
Ibrahimiyah Canal
Wasta
Escape
Desert

PAPER III.

ALTERNATING CURRENTS OF ELECTRICITY;

THEIR

GENERATION, MEASUREMENT, DISTRIBUTION, AND APPLICATION.

BY G. KAPP, M. INST. C.E.

I.—INTRODUCTORY CHAPTER.

HEN we think or speak of electric currents we are accustomed to gard them in the light of material currents, of something which ws along a path formed by the conductor, and has, therefore, a rection. We say that electricity flows along the conductor or rough the conductor from the place of higher to that of lower tential; in the same way that water will flow from the higher to e lower level through a pipe. Such a view is, of course, purely nventional. As a matter of fact we do not know whether it is the sitive electricity that flows in a given direction, or the negative ctricity that flows in the opposite direction, or whether both ctricities flow simultaneously in opposite directions, or whether re is any transfer of electricity through the wire at all. Indeed, ording to modern views, there is merely transfer of energy, but t through the wire, the transfer taking place throughout the ce surrounding the wire. To talk about an electric current flowing through a wire may, therefore, be an unscientific way of expressing our meaning, but it is a very convenient way, and, therefore, generally adopted. Now in adopting this conception of the flow of a certain thing called electricity along a predescribed path, we have also adopted the idea that this flow takes place in a direction which is perfectly well defined in each given case. We have no sense by which we can directly perceive an electric current or note its direction. It is true that if we get a shock we are made aware that a current has passed through us, but no number of shocks will help a man in the slightest degree to an understanding of the real nature of electric currents, nor enable him to determine their direction. We must be content to study, not the currents themselves, but their chemical, thermic, magnetic, and mechanical effects. Amongst other things we must also determine the direction of currents by one or other of these effects. For instance, we know that a wire stretched north-south over a compass needle, and carrying a current, will deflect the needle. If the north-seeking end is deflected to the left or westward, we know by Ampères' rule that the current flows from south to north. Conversely, if the deflection is in the opposite sense, we conclude that the current is from north to south. If the current is obtained from a battery without the intervention of any piece of moving apparatus, such as a reversing key, we notice that the needle once deflected remains in that position as long as the current flows, and we naturally conclude that the current flows continuously in the same direction, that it is, in fact, a "continuous" current. Now suppose you were to notice that the needle, after remaining deflected to the left for a certain time, were to swing over to the right and to remain deflected in that position an equal time, then again swing to the left, and so take alternately these two opposite positions, you would immediately conclude that someone had put a reversing key into your circuit, and was amusing himself by working it at regular intervals. The behaviour of the needle would, in fact, have shown you that you have no longer to do with a continuous current, but that your current has become an alternating current, that is, a current which changes its direction periodically. You will notice that I have assumed that the needle has time to follow each impulse of the current, in other words, that the periodic time of the current is large in comparison with the time of oscillation of the needle. Suppose, however, that I were to work the reversing key so fast that the needle cannot follow the different impulses; in this case it will, of course, remain in its north-south position, and will have become useless as an instrument for the detection of an alternating current. We require an apparatus which will respond far more readily than a sluggish compass needle to the different current impulses which follow each other with great rapidity. To get such an apparatus, let us take an iron diaphragm, and hold near the centre of it a coil of insulated wire forming part of the circuit, or better still, an electromagnet with a laminated core; why the core should be laminated I shall explain later on. For the present it interests us to note that the poles must be in such a position that at least one of them may act on the diaphragm. Thus a ring-shaped magnet which has no free poles would not serve our purpose; a straight bar magnet, however, will do well. Now observe what happens if an alternating current is sent through the coil of this magnet. At the moment of pressing down the key to complete the circuit the battery begins to send a continuous current through the coil and the core begins to get magnetized. The magnetization grows from zero to a maximum, and retains that value until the key is lifted again, when it falls to zero. Now reverse the current and go through the same process. It is obvious that at each reversal of the current the magnetisation must pass through zero, and the end of the core which is presented to the diaphragm will alternately become a north and south pole. The diaphragm will, therefore, be alternately attracted and released, or, in other words, it will vibrate, and if the period of vibration is quick enough, that is, if I manipulate the reversing key very rapidly, a musical note may be produced. Conversely, if I approach an electromagnet to a diaphragm and find that the latter is not permanently attracted, but is set in vibration and emits a musical note, then I conclude that the current which flows through the coil of the electromagnet is an alternating current, and the rapidity of the alternations, or, as it is called, the "frequency" of the current, can be judged from the pitch of the note. In explaining this experiment I have, for the sake of simplicity, assumed that the current is furnished by a battery, and that its alternating character is produced by means of a reversing key. This mechanism is, however, not an essential part of the experiment or of its explanation. The essential part is that the current shall grow from zero to a maximum, and diminish again to zero, then change its direction and grow to a negative maximum, diminish to zero, then become positive again, and so on. Such a current is produced by a certain class of electric machines called "alternators," which will occupy us a good deal during this lecture.

But before entering into this subject I wish to show you experimentally the fact that an alternating current can produce these oscillating or wave-like magnetic effects which I described a moment ago. The apparatus I shall use in my illustrations is extremely simple. I have here a small electromagnet of the kind used in connecting arc lamps to alternating current circuits, and which is technically termed a "choking coil." For a diaphragm I use the bottom of an ordinary biscuit tin, and you will observe that when I approach one end of the choking coil to the biscuit tin there is emitted a sound which can be heard all over the room. The sound is not exactly a clear musical note, because, as might have been expected in a rough-and-ready apparatus of this kind, the elasticity of the diaphragm is by no means perfect. But such as the sound is, it serves quite well to show that the diaphragm is set vibrating by the current, and, in fact, every telephone receiver exemplifies the same action.

The study of alternating currents is greatly facilitated by a rational and simple manner of representing them graphically. There are various ways in which we can so represent not only alternating currents, but any quantity which varies periodically. The most obvious way of representing an alternating current is by drawing a curve, the two co-ordinates of which represent time and the instantaneous current strength. In *Fig.* 1 the time is measured on

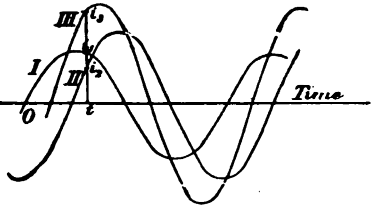


Fig. 1.

the horizontal, and the current strength on the vertical. We thus obtain a wavy line which cuts at regular intervals through the axis of abscissæ. These are the points of reversal when the current strength is maximum, positive where the line lies above, and negative where it lies below the axis. The exact shape of the curve depends on the construction of the machine which produces the alternating current; but I may at once say that in nearly all the theoretical investigations of alternating currents it is assumed that the curve follows, or rather represents, a sine function, and that this assumption is sufficiently near the truth for all practical purposes. All of you know, of course, what a sinusoidal curve is, and I need, therefore, not explain it at length. As, however, the way of plotting a sinusoidal curve brings me to a second method of representing an alternating current graphically, I must say a few words about it. Imagine yourself standing some distance in front of a steam engine in a line with the axis of the cylinder, and looking at the crank pin. The latter will then appear to be moving up and down, making equal excursions to both sides of the centre of the crank shaft. You will, in fact, see the projection of the crank on a vertical, and the length of this projection at any instant is equal to the length of the crank multiplied with the sine of the angle which the crank makes at that instant with the horizontal. The angle is, of course, the product of the angular velocity and the time; and since the angular velocity is constant, you will also obtain a sine curve by plotting the time on the horizontal and the projection of the crank on the vertical. The curve I in *Fig.* 1 has been so obtained. We may, however, save ourselves the trouble of plotting this curve, for we can represent the alternating current more directly by the projection on the vertical of a line OI (*Fig.* 2) revolving with a constant angular speed round the fixed centre O.

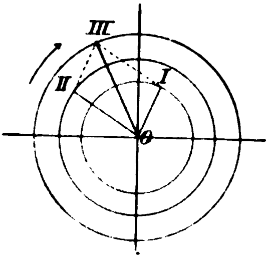


Fig. 2.

The length of OI represents to any convenient scale the maximum value of the current, or the crest of the current wave, and its projection represents its instantaneous value. You see that for half a revolution this value is positive, and for the other half of the revolution it is negative.

In this diagram, which is called a "clock diagram," we must therefore make a projection in order to find the instantaneous value of the current. This is less laborious than the plotting of a sine curve, but it is possible to represent the current in a still more simple way. Those of you who are familiar with Zeuner's valve diagram will immediately see how this can be done. Instead of drawing the circle round O as centre, we draw it passing through O. The diameter of this circle (*Fig.* 3) represents to any convenient scale the maximum value of the current. Then the instantaneous current is given directly by the length of the revolving line between O and the circle. To obtain the negative values of the current, we reproduce the circle on the opposite side; this in the figure is shown dotted.

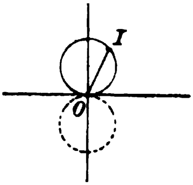


Fig. 3.

To illustrate the use of any of these graphic methods of representing alternating currents, let us suppose that we have to solve the following problem:—We have an iron core wound with two independent coils, each carrying an alternating current. The two currents shall have the same frequency, that is to say, the time which elapses between two succeeding positive maxima or negative maxima shall be the same for both currents, but the maxima in the two currents shall not occur at the same moment. In other words, the phase of one current shall lag behind that of the other, just as in a two-cylinder steam engine one crank lags behind the other. Now the problem we have to solve is: what will be the magnetization of the core at any instant? To find this we must of course know the instantaneous value of the exciting power, or the ampère turns resulting from the action of both currents combined; we must, in fact, find what resultant current acting alone will have the same effect as the two given currents acting together. Let, in *Fig.* 1, the curves I and II represent the two currents, or better still the ampère turns of these currents, then the ampère turns of the resultant current are found by plotting the algebraical sum of the ordinates. Thus we obtain curve III. It is self-evident, and needs, therefore, no elaborate proof, that this curve can also be obtained from *Fig.* 2 if in that figure we draw a parallelogram of currents (precisely in the same way as in mechanics we draw a parallelogram of forces), and use the resultant O III to plot the sine curve. You see that we can combine currents in the same way as mechanical forces. I have proved this for the case that the currents flow in two independent coils, but a glance at *Fig.* 4 will show you that it also holds good if the two currents are sent through the same coil. Here we have two machines, A1 and A2, mechanically coupled, and therefore producing currents of the same frequency. These currents, I and II, flow into one circuit containing a coil C. It is evident that in the circuit BCD there flows only one current, which is the algebraic sum of I and II.

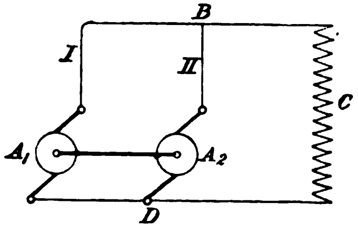


Fig. 4.

Now let us change the arrangement to that shown in *Fig.* 5.

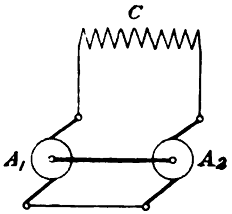


Fig. 5.

Here we have to do with only a single current, for both machines and coil C are coupled in series; but we have to do with two electromotive forces, namely, those of the machines. I assume that the coil C in itself has no electromotive force. In this case also it is self-evident that the current which will be forced through C is due to the algebraic sum of the two electromotive forces, and that all I have said about the determination of the resultant current is directly applicable to that of the resultant electromotive force. In other words, we may use any of the three graphic methods of representing currents also for representing electromotive forces.

These graphic methods of investigation, and especially those based on the clock diagram, are so useful and so simple that I shall employ them frequently in the course of these lectures in preference to analytical methods, and it is, therefore, expedient to familiarize you at the outset with the clock diagram. For this purpose I select, by way of example, a case which is very frequently met with, and which is represented by *Fig.* 6. Lest you should think that this case has merely theoretical importance, I may at once say that a certain deduction which flows naturally from its consideration is of great practical importance in motors driven by multiphase currents, since on it depends the starting torque of such motors. If you compare *Figs.* 5 and 6 you will find that they only differ in this: that an electromagnet S has been substituted for the machine A2. The circuit represented in *Fig.* 6 consists of a machine A, giving an alternating electromotive force, a resistance B, consisting of a bank of glow lamps and an electromagnet S. This electromagnet has a property which is technically called "self-induction," and, before going further, I must briefly explain to you what is meant by self-induction. You know that an electromotive force is set up in a wire whenever the wire cuts across magnetic lines of force. Since the wire must necessarily form part of a closed circuit (for if the circuit were not closed there could be no current), the cutting of lines must be accompanied by an increase or decrease in the number of lines or total induction threading through the circuit, and we may, therefore, also say that whenever the total induction through a circuit changes, there is an electromotive force set up in the circuit which is the

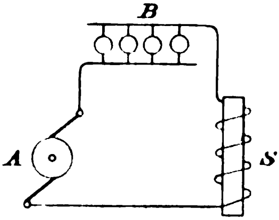


Fig. 6.

greater the more rapid the change. In fact, the rate of change, that is number of lines added or withdrawn per second, multiplied with the number of turns of wire, gives the electromotive force set up in the coil. Going back to *Fig.* 1, we have seen that the curve I represents the current as a function of the time. Suppose there is no other coil wound over the core, then the ordinates of the curve represent to a suitable scale also the exciting power on the core, and it is obvious that the magnetization of the core, or, to speak correctly, the total induction passing through it, will change more or less in accordance with the curve I. If the permeability were constant, the induction would be strictly proportional to the exciting power, and by the selection of a suitable scale the curve representing induction could be made to coincide with the current curve I. Now for low values of the induction, say between zero and 3,000 or 4,000 lines per square centimètre, we may regard the permeability of soft, well-annealed wrought-iron as approximately constant, and if we do not press the induction beyond this point, we may without any great error assume that the current curve I also represents the total induction through the core. For the points where the current passes through zero, and which momentarily interest us the most, the assumption is, of course, quite correct. But if the curve I represents the total induction, then the geometrical tangent to it at any point represents the change of induction in unit time, or, as I said just now, the rate of change of induction at the particular moment represented by the point on the curve. Thus, reading off the time on the horizontal axis, we can, by drawing the tangent to the current curve at the corresponding points, find the rate at which the total induction changes at each moment. I said just now that the rate of change, multiplied with the number of turns in the coil, gives the electromotive force generated at any instant in the coil, and it will now be clear to you that this electromotive force, which we call the "electromotive force of self-induction," must be proportional to the geometrical tangent to the current curve. The steeper this line, the greater is the electromotive force. Thus you see that when the current is either a positive or negative maximum, the tangent is horizontal, and, therefore, at those moments the electromotive force of self-induction is zero. On either side of maximum current it has a definite value, but this value is positive on one side and negative on the other side of maximum current, since the slope of the tangent changes from upward to downward when passing this point. Where the current curve intersects the horizontal axis, the slope of the tangent is evidently greatest, and we, therefore, see that the electromotive force of self-induction is a maximum when the current passes through zero, and it is itself zero when the current is a maximum. This then is, in general terms, the relation between the current curve and the curve giving the electromotive force of self-induction. It remains yet to determine the exact nature of the latter. We have seen that the ordinates of the electromotive force curve are proportional to the geometric tangent drawn to the current curve. Now how do we draw the tangent to the point A, for instance? We draw a straight line through this point, and one very near it, on the current curve. To speak correctly, I should say infinitely near it. At this infinitely near point the current will have increased from i to $i+di$, and the time from t to $t+dt$. The ratio of di to dt is therefore equal to the geometrical tangent at A. But this ratio is the differential quotient of the current in respect to time, and we thus find that the curve giving the electromotive force of self-induction is the first differential of the current curve.

I have up to the present entirely avoided the use of mathematics, but now it becomes necessary to introduce a few simple formulæ. Going back to *Fig.* 2, suppose that the radius OI, the projection of which gives the instantaneous value of the current, makes n complete revolutions per second. Its angular speed is then $\omega = 2\pi n$, and its angular position at the time t is $\alpha = \omega t$, counting the time from the moment that the radius is horizontal. Let I be the length of the radius, which also represents the maximum of the current strength, or crest of the wave, then the instantaneous value of the current at the time t is

$$i = \mathrm{I} \sin \omega t \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1),$$

and the electromotive force of self-induction at that moment is

$$e_s = \mathrm{L}\omega\mathrm{I} \cos \omega t \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2),$$

L being a coefficient which depends on the permeability of the core, the magnetic reluctance of the whole magnetic circuit, and the number of turns in the coil. At the time when the current passes through zero we get the maximum value of the electromotive force, which is

$$\mathrm{E}_s = \mathrm{L}\omega\mathrm{I} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3),$$

and we can also write the equation for the instantaneous value of the electromotive force of self-induction in the form

$$e_s = E_s \cos \omega t,$$

or

$$e_s = -E_s \sin\left(\omega t - \frac{\pi}{2}\right) \text{ .... (4)},$$

from which you will see that this value may also be graphically represented by a sine curve, but lagging behind the current curve by 90 degrees. To obtain the electromotive force curve, which is shown dotted in *Fig.* 7, we must, therefore, imagine a crank OE_s

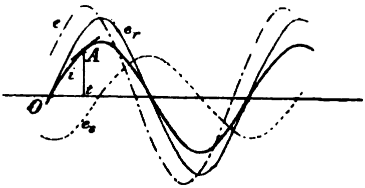


Fig. 7.

rigidly attached to the crank OI (*Fig.* 8) at an angle of 90 degrees, and we must plot the projections of this second crank on the vertical as ordinates in *Fig.* 7. A study of this diagram (*Fig.* 7) will help you materially to an understanding of the phenomena of self-induction. As time progresses, from left to right you see that at first the current is positive and increases. But self-induction opposes the increase, and its electromotive force is, therefore, negative. The dotted curve is below the axis. This opposition becomes fainter as the current approaches its maximum value, since the rate of change of the induction becomes less and less. When the current has reached its positive maximum, the rate of change has become zero, and the opposition of self-induction has vanished. The dotted curve passes through the axis. A moment later, the current is still positive, but is now decreasing. Again self-induction opposes the change; its tendency is to keep the current up at its maximum strength. The electromotive force of self-induction tries to push on the current; it is positive, and the dotted curve rises above the horizontal. This tendency to push on the current increases until the current has become zero, and begins to flow in the reverse direction. The negative current is now opposed by the positive electromotive force of self-induction, but the opposition grows fainter as the current grows stronger, and so the see-saw of pushing on and checking back the current is kept up.

The point which interests us most is what must be the electromotive force given by the machine A, in *Fig.* 6, to produce the current shown by the curve in *Fig.* 7. To simplify our investigation we shall assume that the ohmic resistance of the coil S and machine A is negligible in comparison with the ohmic resistance of the bank of lamps B, or that it is included therein; also that the only part of the circuit having self-induction is the coil S. Then it is immediately obvious that a voltmeter placed across the terminals of this coil will indicate the electromotive force of self-induction, and a voltmeter placed across the lamps will indicate the electromotive force corresponding to the product of current and the resistance of the bank of lamps. But it is not immediately obvious that the sum of these two readings will give us the electromotive force as measured by a voltmeter across the terminals of the machine, and I will show you presently, by theory and by experiment, that this is not the case. Assuming the resistance of the bank of lamps to be a fixed quantity r, it is clear that the instantaneous lamp volts equal the product $r \times i$, and that they can be represented by a sine curve e_r of the same phase as the current curve. In *Fig.* 8 the radius of

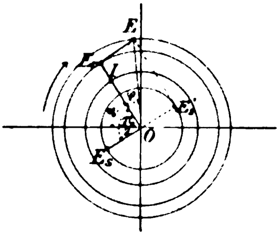


Fig. 8.

maximum lamp volts OE_r must, therefore, coincide with the radius of maximum current OI. The radius of maximnm volts of self-induction is OE_s, and this, as I have already shown, lags behind the current radius by 90 degrees. To find the machine volts at any instant, we must combine the curves e_r and e_s, but remember to take the latter with the opposite sign, for the self-induction opposes the current. This gives us the curve e in *Fig.* 7. To find the machine volts from *Fig.* 8 we have to draw a radius of such length and position that it may be regarded as the resultant of the lamp volts E_r, and an electromotive force diametrically opposed to that of self-induction. We prolong, therefore, the line E_sO beyond O, and make $OE_s' = OE_s$. Completing the parallelogram, we thus find the resultant OE, which gives us the maximum machine volts, or "impressed electromotive force."

The diagram (*Fig.* 8) is very instructive. In the first place, it enables us at once to find an expression for the angle of lag ϕ. You see that the tangent of this angle is given by the ratio of the electromotive force of self-induction to that usefully expended over the lamps. I must here remark that when I speak of electromotive force and current I mean, for the present, always their maximum values. Retaining the notation previously employed, we have, therefore—

$$\tan\phi = \frac{L\omega I}{rI},$$

$$\tan\phi = \frac{L\omega}{r} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (5).$$

Next we can find an expression for the current as a function of the impressed electromotive force and the constants of the circuit. Since the triangle OE_sE is rectangular, we have

$$E^2 = E_r^2 + E_s^2,$$

or, with our previous notation—

$$E^2 = r^2I^2 + L^2\omega^2I^2,$$

$$I = \frac{E}{\sqrt{r^2 + L^2\omega^2}},$$

$$I = \frac{E}{r}\,\frac{1}{\sqrt{1 + \left(\frac{L\omega}{r}\right)^2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (5a).$$

If we had to do with a continuous current, its equation would be $I = \frac{E}{r}$. Since the term under the square root must, under all circumstances, be larger than unity, the current produced by an alternating electromotive force must always be smaller than the current which an equal but continuous electromotive force would produce in the same circuit.

The term $\sqrt{r^2 + L^2\omega^2}$ is called the "impedance" of the circuit, and $L\omega$ its "inductance." As an aid to memory, I reproduce in *Fig.* 9 Dr. Fleming's diagram, in which these terms are recorded. You have seen that Ohm's law is not applicable to alternate current circuits, but if we substitute the impedance for the ohmic resistance, this law becomes applicable.

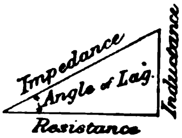


Fig. 9.

I have yet to explain the meaning of the quantity which we called L, and which we introduced in order to take account of the number of turns in the coil, and other properties of the circuit. Of course, most of you will long ago have recognized in this L the usual coefficient of self-induction, but, for the sake of completeness, I must prove this. There are various definitions for the coefficient of self-induction, but the following will serve my purpose:—"The coefficient of self-induction is the ratio between the counter electromotive force in any circuit and the time rate of variation of the current producing it."* In symbols—

$$e_s = L \frac{di}{dt} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(6),$$

and substituting i from equation (1) we have—

$$e_s = L\omega I \cos \omega t \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(2),$$

which is identical with equation (2), and shows that the L we then introduced is indeed the coefficient of self-induction.

II.—MEASUREMENT OF PRESSURE, CURRENT, AND POWER.

When showing you the last experiment, I had occasion to use a voltmeter, and the question we have now to consider is what is the relation between the reading of the instrument and the maximum electromotive force in the circuit. That the reading must be less than the true maximum is obvious, but less by how much ?

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* Sumpner, *The Variation of Coefficients of Induction. Phil. Mag.*, June, 1887, p. 453.

To answer this question we may use the analytical or the geometric method. I give the former in the Appendix, and the latter, which is due to Mr. Blakesley, in *Fig.* 10. A Cardew voltmeter measures not directly volts, but simply the amount of heat developed in its wire per unit of time. The rate at which heat is developed at any instant is the product of the instantaneous current and the instantaneous volts; but as the current passing through the wire is proportional to the volts, the rate at which heat is developed is proportional to the square of the instantaneous volts, that is, to O_e squared in *Fig.* 10, if by OE we represent the maximum volts. Now, to find the general effect of a large number of succeeding instantaneous voltages on the voltmeter, we have to draw the projections O_e of OE for a large number of positions. Let us take these positions in pairs, such as OE and OE′, with an angular interval of 90 degrees between them. It is evident that for each such pair the sum of the squares of the projections is equal to the square of the maximum voltage, and that the mean voltage is $\frac{1}{2} E^2$. This is independent of the actual position of each pair, and is, therefore, the mean value of all the possible pairs. The volts read on the Cardew (or any other instrument, the action of which depends on the square of the voltage) must, therefore, be multiplied by the square root of 2 in order to get the maximum volts; or in symbols, if by e we represent the volts shown by the instrument, and by E the maximum volts—

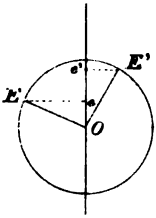


Fig. 10.

$$e = \frac{E}{\sqrt{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(7).$$

At the Paris Congress, in 1889, it has been decided to call e the "effective" volts.

The same reasoning which I have here applied to the measurement of pressure can also be applied to the measurement of current, provided that the measuring instrument is of a kind in which the movable part is subjected to a force varying as the square of the current. Thus a Siemens' dynamometer, a Thompson ampère-balance, and similar instruments, will indicate the "effective current"—

$$i = \frac{I}{\sqrt{2}} \quad \text{..........................(8).}$$

It is important to observe that this is not the same thing as the mean or average current. To understand the distinction, let us first settle what quantity we call the mean current. Imagine an alternating current commutated each time it passes through zero, and let the unidirected but pulsating current thus obtained pass through an electrolytic apparatus, say, for instance, a copper voltameter. The weight of copper thrown down in unit time is a measure of the "mean strength" of our pulsating current. The mean current strength thus defined is about 90 per cent. of the effective current strength, so that to get the true mean current we must multiply the reading of the Siemens' dynamometer by 90. The proof for this is given in the Appendix.

We have seen that the measurement of pressure and current is a very simple matter; but the measurement of power, to which I must next draw your attention, is not quite so simple as with continuous currents. You know that if we wish to determine the power given to a circuit by a continuous current we have merely to observe the ampères and the volts, and multiply them to get the watts. If we divide the watts by 746, we get the result in horse-power. With alternating currents this is not quite so simple, and if we were to compute the power in this manner, our results would be generally too large, and never too small. The reason, of course, is that the instant of maximum ampères does generally not coincide with the instant of maximum volts. To get the true power we must integrate the product of the instantaneous volts and ampères over a complete cycle, and divide by the time required to perform the complete cycle. To do this here would require more time than we have at our disposal, but you will find the matter treated in the Appendix analytically. Here I treat it graphically, using again Mr. Blakesley's method. In *Fig.* 11 OE and OI represent electromotive force and current at any given moment, OE′ and OI′ their position a quarter period later. The current lags behind the electromotive force by the angle ϕ. Using small letters for the projections of E and I on the vertical, the mean value of the power for the two positions is obviously $ei + e'i' \div 2$.

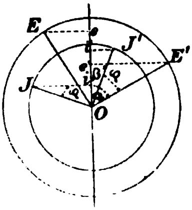


Fig. 11.

From the diagram, the following equations are obvious:—

$$ei = \mathrm{EI} \sin \alpha \sin \beta.$$
$$e'i' = \mathrm{EI} \cos \alpha \cos \beta.$$

Combining these we find the mean power—

$$w = \frac{\mathrm{EI}}{2} (\cos \alpha \cos \beta + \sin \alpha \sin \beta).$$

$$w = \frac{\mathrm{EI}}{2} \cos (\alpha - \beta).$$

$$w = \frac{\mathrm{EI}}{2} \cos \varphi \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (9),$$

and this is the same for every pair the position of which differs by 90 degrees. It is, therefore, the true equivalent power.

Since $e = \frac{\mathrm{E}}{\sqrt{2}}$ and $i = \frac{\mathrm{I}}{\sqrt{2}}$, we have also—

$$w = ei \cos \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (10),$$

where e and i are the volt and ampère readings, as obtained by our usual instruments.

You see we have first to determine the "apparent" watts as if we had to do with a continuous current, and then to get the true watts we must multiply by the cosine of the angle of lag. It is, however, not always easy to determine the angle of lag, and to avoid the labour and possible errors of such a determination, various instruments have been invented for the direct measurement of power, which are called wattmeters. The best-known form of wattmeter is constructed similarly to a Siemens' dynamometer. The fixed coil, containing a few turns of thick wire, is connected in series with the main circuit, and the movable or suspended coil, containing many turns of fine wire, is connected as a shunt to the main circuit. A non-inductive resistance is put in circuit with the movable coil to reduce the self-induction of the shunt circuit. (For theory of wattmeter, and corrections to be applied, see Appendix V.).

III.—CONDITION OF MAXIMUM POWER.

It is important to investigate the conditions under which we can obtain a maximum of power in a given circuit. This is the deduction of which I have spoken a little while ago as flowing naturally from the investigation of the case represented by *Fig.* 6. Here we have an alternating current machine, a self-induction, and a bank of lamps. The self-induction we cannot diminish, and the machine volts we cannot increase. How must we manage our lamps to get a maximum of power into them, and, therefore, to get a maximum of light out of them? Without entering into any lengthy mathematical investigation, you can see at once that the ohmic resistance of the bank of lamps will have a great deal to do with the amount of power usefully expended. If the resistance is very high, the lamps will get very nearly the whole of the machine volts, but then the current will be small. If, on the other hand, we lower the resistance too much in our desire to get a large current, we shall have to sacrifice nearly the whole of the pressure, since the self-induction, which now is fed by a large current, will choke back most of the available voltage. You see that either too little or too much resistance is bad, and we have to find that resistance which will give us the best effect. This will be the case when the volts over the lamps equal the volts over the self-induction, either being about 70 per cent. of the machine volts. I give the analytical proof in Appendix IV., and the geometric proof by means of the clock diagram (*Fig.* 12). Let, in this figure, the circle represent the given machine volts, and let OE_s be the volts of self-induction corresponding to the current OI. Then the tangent of the angle at I is obviously equal to $L\omega$ (by equation (3)). The power given to the lamps is (by equation (9)) $w = \frac{1}{2}$ IE cos ϕ, that is, the projection of the volt radius on the current radius, multiplied by the current, and divided by 2. But the projection of the machine volts OE gives us the lamp volts OE_r and the current is proportional to the volts of self-induction (see triangle

OIE$_s$), so that we can also say the power is proportional to the product of lamp volts and self-induction volts (that is, to the shaded area in *Fig.* 12), and our problem can also be stated in these terms :—Find that position of OE for which the shaded area becomes a maximum. Obviously this will be the case when the line OE forms an angle of 45 degrees with the horizontal; in other words, when the current lags by one-eighth of a period behind the impressed electromotive force, and when the volts measured over the self-induction equal the lamp volts. This condition will be fulfilled when the resistance of the bank of lamps is

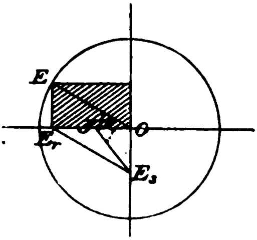


Fig. 12.

$$r = L\omega.$$

In order to simplify the explanation, I have assumed that in *Fig.* 6 the resistance, the self-induction, and the seat or source of electromotive force, are different and distinct parts of the circuit. This was, however, not essential. We could, for instance, assume the self-induction to be part and parcel of the machine, or even of the bank of lamps, and yet our result would have been the same. Nay, more. Suppose we take away the lamps and put in their place a series wound dynamo with well laminated field magnets. Whichever way the current is sent through the machine it will always revolve in the same direction, and the alternating current must, therefore, set it in motion. Now, imagine the period of the current very long, in fact, so long that the electromotive force of self-induction of the magnet coils and armature may be neglected. Then the only force opposing the current in its flow through the armature is the counter electromotive force developed by the latter, which, at constant speed of rotation, is proportional to the field strength. Now, if we do not excite the magnets strongly (and on account of hysteresis and other losses it is advisable to work with a low induction), we may consider the field strength to be proportional to the current, so that the only force which opposes the current will in this case be at all times proportional to the current strength. It will, in fact, be the same kind of opposition as is produced by ohmic resistance, but with this difference, that, instead of converting the electric power into heat in the lamps, we convert it into mechanical power, which may be taken off the spindle of the motor. So far the motor, although supplied with an alternating current of very long period, will work exactly as if it were joined to a continous current circuit. But now let us increase the frequency of a current, that is to say, let us shorten the period and have more and more current waves per second. This will add to the counter electromotive force of the motor (which is useful, because accompanied by the giving out of mechanical power) another electromotive force which is entirely useless, namely, the electromotive force of self-induction. This must considerably decrease the power obtainable from the motor, first, because the current strength has been decreased by its action; and, secondly, because with the electromotive force of self-induction now having become a large quantity, a considerable lag of the current behind the machine volts must take place. A few years ago I experimented with a motor of this kind, and found that the power obtainable from the machine when coupled to an alternating current circuit was only about one-sixth its normal power on a continuous current circuit. In this motor the self-induction was far too large as compared with its counter electromotive force. By our rule, the electromotive force of self-induction should have been equal to the counter electromotive force. In this case the motor would give about 70 per cent. of the power it could develop with a continuous current. If it were possible to make motors which fulfill the condition I have explained, it would be a very easy and practical solution of the problem how to make the existing lighting stations which supply alternating current available for the distribution of power, but I doubt very much the possibility of this solution of the problem. The self-induction of such a motor must always be enormously high, but a way in which we can at least approach the best condition of working is by lowering the frequency and increasing the rotary speed of the motor. I have devoted some time to this method of working alternate current motors, because it has already acquired practical importance, not indeed in the regular working of these machines, but in the starting of them. The Ganz motor is started without a load, as if it were an ordinary continuous current machine, and after it has acquired a certain speed it suddenly begins to work as an alternating current machine. When in this condition the load can be thrown on, and the machine will even stand a certain amount of excess load.

IV.—ALTERNATING CURRENT MACHINES.

When discussing the electromotive force of self-induction, we have seen that this is produced by the change in the total induction passing through the electric circuit, or, which comes to the same thing, by the cutting of wires across magnetic lines of force. In a choking coil the magnetization changes, and we thus obtain an electromotive force without the necessity of moving the wire; but when the strength of the field is a constant, and its position in space remains the same, then we must move the wire in order to get an electromotive force, and this is precisely what we do in our alternating current machines or "alternators." The most simple conceivable form of alternator is shown in *Fig.* 13. Here we make use of the vertical component of the earth's magnetic field, and the electromotive force is a maximum when the wire is either at its highest or lowest position (crank vertical), and when the wire is in the extreme right or left position (crank horizontal) the electromotive force is zero. The apparatus shown in *Fig.* 13 is, in fact, simply a mechanical model of the clock diagram. In this figure the bearings and standards supporting them are represented as forming the terminals; and wires attached to them as shown, and led to a solenoidal electromagnet, will energize the latter. Provided the strength of the field were sufficiently great, we could, when the crank is rapidly rotated

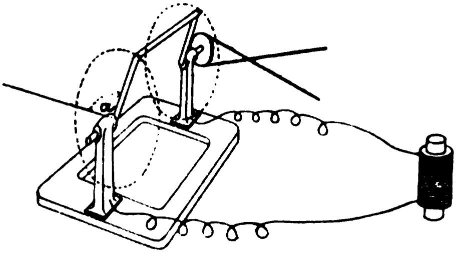


Fig. 13.

by a cord and pulley, produce with the electromagnet the effects I have shown you when the coil was connected to a transformer. But the strength of the field provided by the earth is not nearly sufficient. We must resort to an artificial field produced by electromagnets, such, for instance, as is shown in *Fig.* 14, where NS are the polar surfaces of two electromagnets, between which a coil C is rotated. If the polar surfaces extend some distance beyond the diameter D of the coil, we may assume that the field within the space swept by the coil is uniform, and then the number of lines or total induction passing through the coil at any instant will be proportional to the sine of the angle the coil forms with the vertical.

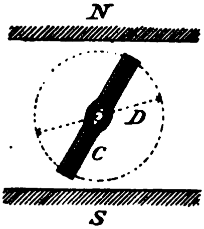


Fig. 14.

The electromotive force will then be proportional to the sine of the angle the coil forms with the horizontal. Calling H the field intensity, l the length of the coil, v the velocity, and τ the number of wires counted on both sides, the maximum electromotive force in C.G.S. measure, when the coil is vertical, will be

$$Hvl\tau.$$

It is convenient to introduce instead of H, the number of lines per square centimètre, the total induction F passing through the coil; and instead of the linear velocity, the number of revolutions per second, which, in this case, where we have to do with a two-pole machine, is equal to the frequency n. A simple algebraical operation, which need not be reproduced here, gives us the following formula for the maximum electromotive force :—

$$E = 2\pi n F \frac{\tau}{2}.$$

The effective electromotive force is obtained by dividing this expression by the square root of two, and if we wish to get the electromotive force in volts we multiply by 10 to the power of minus 8, or in symbols—

$$e = 2{\cdot}22, \mathrm{F}\tau n \; 10^{-8} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (11).$$

Now suppose we remove the armature shown in *Fig.* 14, and replace it by one wound to give a continuous current. We use the same total length of wire, but spread the turns evenly all round the circle and put on a commutator. As you know, the electromotive force of this machine will now be

$$e = \mathrm{F}\tau n \; 10^{-8}.$$

The current flows through the armature in two parallel circuits, and if we allow the same current density in the armature wires, we shall obtain a continuous current of twice the strength of the alternating current. On the other hand, the alternating current will have 2·22 times the voltage, so that the output of the alternator will be 11 per cent. greater than that of the continuous current machine. In the alternator we save the commutator, and you see, therefore, that for equal output the alternator is a cheaper and lighter machine than the continuous current dynamo.

The alternator shown in *Fig.* 14 is, however, not the kind of machine used in practice. I have only chosen it as a simple example to show the relation between continuous and alternate current machines. In practice the latter are made with a number of poles in order to bring down the speed to reasonable limits, and the wire is not bunched together, but is spread more or less over the surface of the armature. There is also this difference: that the poles of the field surround the armature more closely, and consequently the transition from a north to a south field is more abrupt than in *Fig.* 14. Notwithstanding these differences, the electromotive force of alternators as practically made is very nearly that given in Formula (11), and the comparison of weight and cost which we found just now holds good for the machines as actually built.

You will see from the diagram on the wall that alternators are all characterized by two main features—a corona or ring of magnet poles and a ring of armature coils, either one or the other being movable. The particular shape of the poles, their arrangement mechanically, the method of winding the armature coils, and many other details, may be changed in many ways, but the main features remain the same.

For purposes of study it is convenient to imagine the pole ring and the armature ring cut open and spread into straight lines. We need then only consider one coil and two or three poles, as shown in *Fig.* 15. The electromotive force at any moment is obviously proportional to the number of wires which happen at that moment to be covered by one or both poles, care being taken to count, in the latter case, the difference in the number of wires, since the poles act differentially. We can thus plot a curve giving the resultant electromotive force as a function of the position of the coil in front of the poles, and since at constant speed this position changes proportionately with the time, the curve also gives us the electromotive force as a function of the time. Now you will easily see that the shape of this curve depends on the width of the poles and length of coil. It also depends on their relative shape. But not to complicate our investigation too much, I assume that both poles and coils are rectangular, which in machines of the Westinghouse and Lowrie Hall type is strictly, and in most other machines is approximately, true. As an extreme case regarding the width of poles, we may

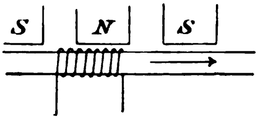


Fig. 15.

take a machine in which the north and south poles are placed so close as almost to touch each other. In this case the width of poles is equal to their distance or pitch. Machines with alternate poles set so closely are not made; but the Mordey, in which poles of the same sign, separated by equal intervals of blank or neutral spaces, is a practical illustration of the same principle. Suppose now that we put into such a field an armature, the whole surface of which is covered by coils, then the length of each coil must also equal the pitch, and there will then be only one position, namely, that in which the centre of the coil coincides with the centre of the pole, when all the wires in the coil are producing electromotive force in the same direction. In every other position the electromotive force in one part of the coil is opposed to that in the other part. In such a machine the wire is not used to the greatest advantage, and the electromotive force curve becomes a zigzag line as shown in *Fig.* 16. The question which interests us most is that of the effective electromotive force represented by this curve. Experimentally we can, of course, easily determine it. We need only connect a Cardew voltmeter to the terminals of the coil and take a reading; but it is important to know beforehand, that is, before the machine is built, what volts we may expect to get. Consider for a moment what it really is that the voltmeter measures. It is the amount of heat developed per second in its wire. With the quick alternations produced by the machine there is no time for the wire to change its temperature, and its resistance is, therefore, constant. The amount of heat dissipated per second is then the square of the effective volts divided by the resistance, and this is also equal to the integral of the square of the instantaneous volts multiplied by the differential of the time, the integration being extended over one second; or we may extend the integration only over the time occupied by one half period and divide the result by that time; this will also give us the heat per second. But as we want to know the volts and not the heat generated per second, we need not concern ourselves about the resistance of the voltmeter wire at all, and simply take the square root of the integral e^2dt; this gives the effective volts. To do this graphically, as shown in *Fig.* 16, scale the ordinates of the zigzag line, square the readings, and plot to an arbitrary scale the result. Thus we obtain the tent-like figure shown in the diagram, the area of which is the integral, or, to speak quite correctly, is proportional to the integral of square of instantaneous volts and time. The height of a rectangle, of equal base and area, is the square of the effective volts. It is thus possible to determine beforehand, for any given arrangement of field poles and armature coils, what the effective volts will be, and, roughly speaking, the larger the shaded area, the higher will be the voltage of the machine. For instance, in *Fig* 17, I have assumed that the field of *Fig.* 16 has been retained but

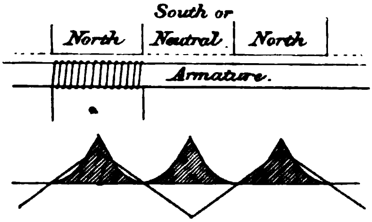


Fig. 16.

that the armature coils have been made only half the length for the same number of turns. Only half the surface of the armature is now covered by wire, and the maximum electromotive force is maintained for a quarter period instead of being momentary as before. This gives a trapezoidal line for the electromotive force curve, and

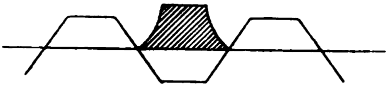


Fig. 17.

the shaded area is now considerably larger than before. Let us now go back to the first armature, and run it in a field the poles of which are only half the width, the total induction being, however, the same. Here again we get a trapezoidal electromotive force curve (*Fig.* 18), and the same voltage as in *Fig.* 17. If we now shorten

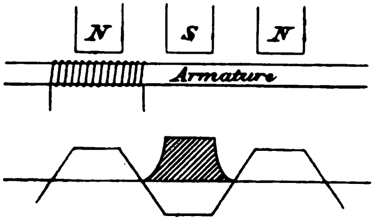


Fig. 18.

the coils, we come back to the zigzag lines, but the peaks are higher (*Fig.* 19), and the voltage, as shown by the shaded area, is again increased. The arrangement shown in *Fig.* 19 is that usually met

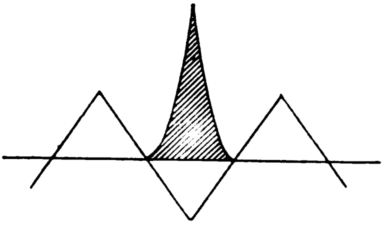


Fig. 19.

with in modern alternators, but owing to the fringe of lines at the corners of the pole-pieces, the electromotive force curve is not quite as sharp as here shown. The peak is rounded off and the sides are more wavy, the curve approaching, in fact, very nearly to a true sine curve. Leaving, however, such refinements aside, it is easy to work out an expression for the effective volts in each given case, either graphically by the use of such diagrams as *Figs.* 16 to 19, or analytically. The operation is somewhat laborious, but in no sense difficult, and it would be useless to burden this lecture with it. I will merely state the results. If you will refer back to Formula (11) you will see that the effective electromotive force of a simple coil, revolving in a uniform field, is given by the product of a constant (in this case 2·22), the total induction or number of lines emanating from one field pole, the number of wires counted on both sides of the coil, and the frequency. If, instead of a machine with two poles and one coil, we had made a machine with 10 poles and five coils, coupled in series, the electromotive force of each coil would have been five times as great; if the machine at 20 poles and 10 coils it would have been 10 times as great, and so on. You see, therefore, that the electromotive force is simply proportional to the total number of wires coupled in series, and to the number of pairs of poles, and Formula (11) is right for a machine with any number of poles. The same kind of formula is also correct for any arrangement of poles and coils, but the coefficient is different in each case. This coefficient is really the ratio of the electromotive force of the alternator to that of a continuous current dynamo of equal weight and arrangement of field and armature. If the machine has r pairs of poles, and runs at a speed of N revolutions per minute, its electromotive force of a continuous current machine is

$$e = pFr\frac{N}{60}10^{-8} \quad \dots\dots\dots\dots (12),$$

or if by Z we denote field strength in English measure—

$$e = pZrN10^{-6} \quad \dots\dots\dots\dots (13).$$

Let K be the coefficient which depends on the shape of poles and armature coils, then the electromotive force of our alternator is

$$e = KpFr\frac{N}{60}10^{-8} \quad \dots\dots\dots\dots (14).$$

$$e = KpZrN10^{-6} \quad \dots\dots\dots\dots (15).$$

$$e = KFrn10^{-8} \quad \dots\dots\dots\dots (16).$$

To find the electromotive force we must, therefore, determine the coefficient k for each case, and, as I have already said, this is not a difficult mathematical problem. The result for the cases I have brought before you is as follows :—

(1). If machine gives a strictly sinusoidal electromotive force .. .. $k = 2\cdot22$
(2). Width of poles equal to pitch, and length of coils equal to pitch ... ... $= 1\cdot160$
(3). Width of poles equal to pitch, and length of coils equal to half the pitch ... ... $= 1\cdot635$
(4). Width of poles equal to half the pitch, and length of coils equal to pitch ... ... $= 1\cdot635$
(5). Width of poles equal to half the pitch, and length of coils equal to half the pitch ... ... $= 2\cdot300$

If you compare the first and last line of this table, you will find that there is only 3½ per cent. difference between the two coefficients. The last line refers to machines as actually built, and the first line to ideal machines having a true sinusoidal electromotive force curve. You see that, as far as the effective electromotive force is concerned, the assumption that ordinary commercial alternators follow the sine law is practically correct.

V.—Mechanical Construction of Alternators.

Speaking generally, we may say that the constructive requirements and the points to which particular attention must be paid in designing alternators are very much the same as obtain in dynamos, but there may be certain differences. In the first place, the armature of a dynamo is, on account of its commutator and brushes, necessarily more complicated than that of an alternator. On the other hand, the field is simpler. The majority of dynamos are made for low or moderate voltage, whilst alternators are generally made for high voltage. This requires greater care in the insulation, and compels us to avoid certain methods of winding, which for a 100-volt dynamo are quite admissible. In the design of both kinds of machine we must pay attention to eddy currents and hysteresis, but in alternators these disturbing and injurious effects are far more serious than in dynamos. The reason is that both the wire and the iron, if the armature has an iron core, are subjected to a more rapid reversal of induction. Special precautions must therefore be adopted. The core must be well laminated, and the conductor should not exceed a certain section. What that maximum section should be depends, of course, on the general design of the alternator, but we may take it roughly that, where round wire is used, its diameter should not exceed 140 mils., and where strip is used its thickness should not be more than 100 mils. Another and very effectual cure for eddy currents is to embed the conductor entirely in iron, an arrangement which has been first proposed by Wenstroem, and has been lagely used by Brown, the latest example being his large three-phase current alternator at Lauffen. The conductor is a solid copper rod of about 1½ inches in diameter, threaded through holes in the armature core. A conductor of that size, if placed on the surface of an armature where it is subjected to some 80 field reversals per second, would get hot in a few minutes, yet, arranged as it is in Mr. Brown's "three-phaser," it keeps perfectly cool. It is the fact that the conductor is surrounded on all sides by iron which produces this result. A still more striking illustration of the effect of iron in preventing eddy currents is Thomson's welding machine. Here we have a solid conductor of many square inches in area, in which the welding current is generated. But this conductor is the secondary circuit of a transformer, and is surrounded by the iron of the transformer. Professor Thomson's explanation of the fact that in all such cases eddy currents are avoided is that the speed at which the wire cuts through the lines of force is much greater than its speed of motion, that, in fact, the lines at first yield and, so to say, stretch, but finally, when the tension becomes too great, snap suddenly past the wire. Thus all parts of the wire are cut at almost the same instant by the lines of force, and this leaves no time during which the differences of electromotive force, and, therefore, eddy currents, could be developed in the wire. I, personally, do not feel competent to either confirm or refute this explanation, but coming from so high an authority as Professor Elihu Thomson, am satisfied to accept it. That wires embedded in iron, or surrounded on all sides by iron, are nearly free from eddy currents is, however, an undoubted fact.

From what I have here said you will see that in one way or another we can avoid, or at least greatly reduce, the loss of power by eddy currents in alternators. Now let us see whether this is also the case with the other source of loss, namely, "hysteresis." Under this term we comprise a certain phenomenon, first investigated by Ewing, and which may be popularly described as "magnetic friction." The lines of force in being forcibly dragged through the iron core of the armature continually change its magnetization, and the core, even if most carefully laminated, so as to avoid eddy currents, still becomes hot if revolved in an excited field. It is at once obvious that the power thus wasted will be the greater the more rapid the reversal of magnetization, and the greater its amount. This loss takes place in dynamos as well as in alternators, but to a different extent. In a dynamo the reversal is comparatively slow. Take, for instance, a two-pole machine, running at 600 revolutions per minute, or 10 revolutions per second. The whole mass of the armature core undergoes, therefore, in every second ten complete cycles of magnetic change. But in modern alternators the change is about ten times as rapid, the frequency being 100. If we allowed the same induction, that is, number of lines per square centimètre of core section, the alternator would waste ten times the power, and this would, of course, be inadmissible. There is only one way in which we can reduce the waste of power, and that is by adopting a lower induction. Thus, whilst in dynamos the induction ranges from 14,000 to 20,000 lines per square centimètre, it is only about 5,000 in alternators. The exact induction at which it is best to work varies, of course, with the type and size of machine, and as every design is a compromise, you must not consider the 5,000 as a hard-and-fast rule. To enable you, however, to deal with each given case on its own merits, I give in *Fig.* 20 a curve showing the loss of

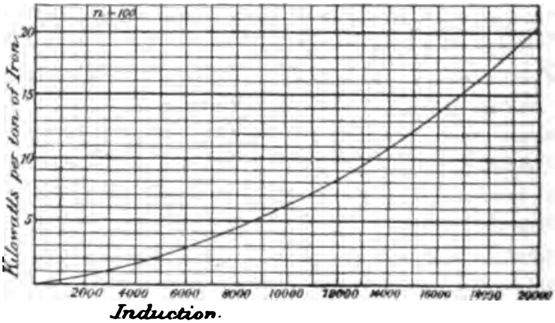


Fig. 20.

power by hysteresis per ton of iron when the frequency is 100. The induction is measured on the horizontal, and the power (in kilowatts) on the vertical. This curve has been compiled from the experimental results of Professor Ewing. I may incidentally mention that this curve is approximately represented by the equation—

$$\text{Power} = 180 \left(\frac{B}{1000}\right)^{1 \cdot 55} \ldots\ldots\ldots\ldots\ldots\ldots (17),$$

or if the induction B is given in English lines per square inch—

$$\text{Power} = \cdot 160\ B^{1 \cdot 55} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (18).$$

The power is given in kilowatts per ton of iron when the frequency is 100 complete cycles per second. For a different frequency the power is proportionally altered.

There being necessarily always some waste of power, if the armature has an iron core it was natural that inventors should turn their attention to the construction of an alternator with a coreless armature. In fact, the Meritens machine, which was one of the first commercial alternators, and is used to this day in lighthouse work, has no iron in the armature. Then there comes the Siemens, also without iron, the Ferranti, and the Mordey. In all these machines the loss by hysteresis is avoided, and if this were the only consideration, they would undoubtedly be better than their rivals with iron-cored armatures. But, as I have said before, every design is a compromise, and it is quite possible that the machine with iron in its armature is as good a compromise as one without iron. The fact that the majority of American machines, all the German machines, including those now made by Messrs. Siemens and Halske, and a good half of the English machines, have iron-cored armatures, is in itself sufficient proof that the hysteresis loss is not an insurmountable obstacle. There are especially two points in favour of using iron. The first is that we are thereby enabled to give the armature greater mechanical strength than can be done in machines where the armature coils are attached singly and held by insulating material. The second is that the presence of iron tends to diminish the magnetic resistance of the air-gap, and thus saves exciting energy. In Mr. Brown's three-phaser, for instance, the total exciting energy does not amount to more than ½ per cent. of the total power.

I have, a moment ago, spoken of the differences between alternators and dynamos from an electrical and mechanical point of view. There remains yet to notice an important point of difference, namely, the absence in alternators of a commutator and brushes. You all know that these are the most delicate parts of a dynamo, and although in modern machines of moderate voltage these parts are perfectly reliable and easily handled, the case is different when we attempt to build dynamos for 1,000 or 2,000 or more volts. We encounter then difficulties which are absent from alternators, and it is mainly on this account that engineers who have to design power transmission schemes over long distances are beginning to turn their attention to some forms of alternator as the most certain means of solving such problems. I shall have something more to say on this subject in the third lecture. For the present I must limit my remarks to the machines as required for lighting.

VI.—DESCRIPTION OF SOME ALTERNATORS.

In the limited time at my disposal it would be impossible for me to give you anything like an exhaustive account of the various machines now in use. I shall, therefore, only describe a few of them as being representative examples.

1. *The Ferranti Alternator.*—The field magnets are wrought-iron bars of trapezoidal section (*Fig.* 21), cast into massive yoke rings, which can be drawn apart at right angles to the shaft, so as to expose the armature for examination and repair. The latter is of disc pattern, and the coils are inserted in pairs. The conductor is a corrugated copper strip wound with a strip of vulcanized fibre of equal width upon a laminated brass core. The conductor is thus insulated from the core, and the latter is insulated from the supporting ring. This double insulation is an important feature of the machine. The core is held in gun-metal cheeks, which are provided with side wings for ventilation. The attachment of each pair of cheeks to

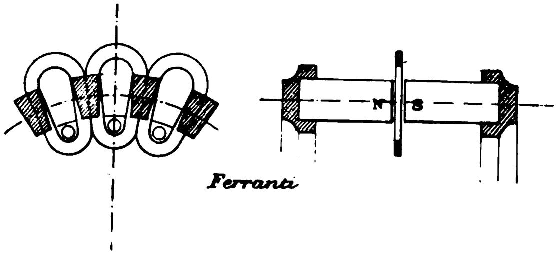


Fig. 21.

the supporting ring is by means of a shank passing through insulating washers into a cavity in the ring, and secured by a nut. The cavity is cast out with sulphur. To avoid too great a loss by eddy currents the conductor is made very thin; the winding is split up into two, four, or more parallel circuits. I may here incidentally mention that where an armature winding is thus split up, great care must be taken to have all the magnets of equal strength, as otherwise there would be created within the armature differential currents, which would waste far more power than the eddy currents, which the arrangement was intended to avoid. The Ferranti machines now working at Deptford are giving an electromotive force of 10,000 volts, and to prevent flashing over to the magnets the latter are provided with double caps of ebonite.

2. *The Mordey Alternator.*—This is also a coreless machine of the disc pattern, but the armature is fixed whilst the magnets revolve. The armature coils (*Fig.* 22) are wedge-shaped, and the conductor is a thin copper strip wound on a slate core, the layers being separated, as in the Ferranti coils, by a thin strip of insulating material. The attachment is made at the outer and wider end of the coil to a gun-metal supporting ring. The magnets are of cast-iron, and so shaped as to require only one coil C of exciting wire. This is wound on a central cylindrical part *y*, to both sides of which are pole pieces of peculiar star-like form. Thus the poles on one side of the armature are all of the same sign, and those on the other side are of the opposite sign, the lines of force passing from N to S at right angles through the surface of the armature, and all in the same direction. There is thus, properly speaking, no reversal of magnetization, but merely a change from full induction when a wire is between opposite

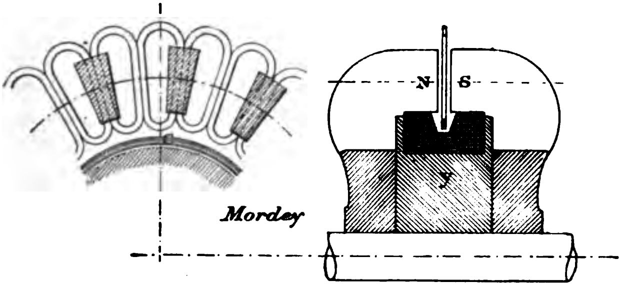


Fig. 22.

poles, to no induction when it is between neighbouring poles, and the general effect is the same as if we had half the field strength alternating. To apply our formulæ for the electromotive force of this machine we must, therefore, introduce not the whole field strength F or Z, but half its real value.

3. *The Westinghouse Alternator.*—As a good example of an alternator, the armature of which contains iron, we may take the Westinghouse machine, which, in its important details, is very similar to the Thomson-Housten. The armature is cylindrical (*Fig.* 23), and is covered by link-shaped coils, with the wires parallel to the shaft, the rounded ends of the coils C being bent inwards, and secured to the end faces of the armature core. In the Thomson-Housten machines the coil ends are not turned inwards. The field magnets NS are set radially outside the armature, and their outer ends are connected by a yoke ring Y. According to our theory, the best arrangement as regards width of coil is half the pitch, which means that the central space of the coil should have the same width as the magnet, but Professor Thomson, when experimenting with various coils, found that a coil having a slightly smaller internal space gave a higher electromotive force when the machine was working under full load. His explanation is that the current in the armature wires alters the original magnetization of the field, tending to concentrate the lines towards the leaving edge of the pole piece, and thus produces a more intense but narrower field. The inner space of the coils, which is free from winding, should, therefore, also be made narrower.

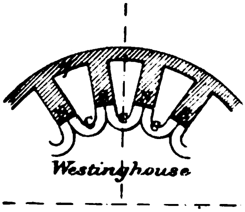


Fig. 23.

4. *The Kapp Alternator.*—In this machine (*Fig.* 24) the armature is of the disc pattern, and contains an iron core A made by coiling a strip of thin charcoal-iron with a strip of paper upon a supporting ring. The coils are wound transversely round the core. The field magnets in the larger machines are of wrought-iron, with expanded

pole shoes, and are set parallel to the shaft on both sides of the armature core, presenting the same poles NN SS on opposite sides. The outer ends of the magnets are joined by cast-iron yokes YY. Owing to the angular position of the pole shoes, each wire does not enter the field simultaneously over its whole length, but the entry is a little more gradual, whereby the sharp peaks in the line of electromotive force (*Fig.* 19) are toned down, and the curve is made to approach a sinusoidal form. The current is collected by rubbing contacts from insulated rings, which are set on opposite sides of the armature, that is, so far apart that it is impossible for a man to touch both simultaneously. The coefficient K for this machine varies between 2·3 and 2·7, according to the particular design chosen.

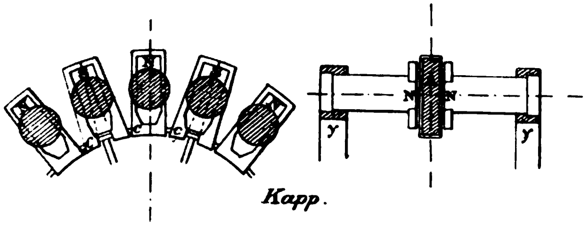


Fig. 24.

5. *The Kingdon Alternator.*—In all the machines described up to here, the wire, either on the field or on the armature, is in motion, but in the Kingdon machine all the wires are at rest, the only revolving part being an armature containing no wire. The machine consists of a laminated iron cylinder, with radial teeth projecting inwards, and the armature and field coils are wound over alternate teeth. The revolving part is a wheel provided with half as many laminated iron keepers as there are teeth in the stationary part, and these keepers are so arranged as to bridge magnetically neighbouring teeth. Thus the teeth over which the armature coils are wound become alternately parts of a positive and negative magnetic circuit, and an alternating current is produced.

6. *The Kennedy Alternator.*—Mr. Kennedy has further developed this idea, mainly by reducing the number of armature and field coils, and avoiding the generation of an alternating current in the latter. The machine (*Fig.* 25) has two armature and two field coils wound in pairs, and placed into recesses in a skeleton frame of soft iron laminated bars. There are two keeper wheels on the spindle, but

stepped in relation to each other by half a period, so that when the keepers of one wheel have completely closed the magnetic circuits around one pair of field and armature coils, the keepers of the other wheel are midway between the fixed bars, and the magnetic circuits around the second pair of armature and field coils are interrupted. The electromotive force in those coils is at that moment zero, and it is also zero in the other coils, through which the induction is a maximum. Half a period later the induction becomes a maximum in the second, and zero in the first pair of coils, and this change of induction produces an alternating electromotive force in all the coils. Now it is obviously possible to couple the two field coils in series, and in such way that the electromotive force created in one is opposed to that in the other, and thus to neutralize the reaction of the keeper on the exciting circuit. The exciting dynamo has then merely to overcome the ohmic resistance of the two coils, as in any other machine. The two armature coils may be coupled in series or parallel.

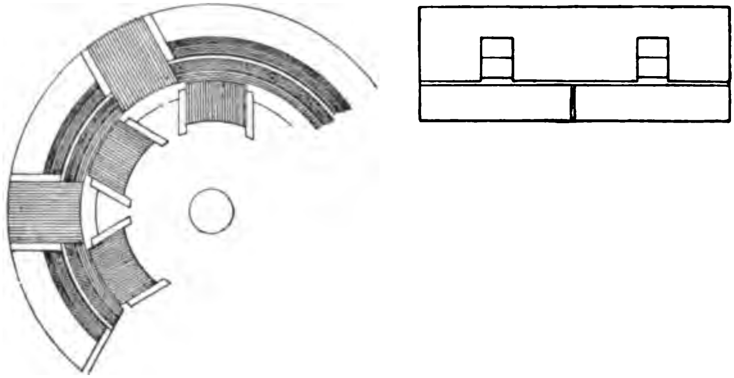


Fig. 25.

VI.—TRANSFORMERS.

I have already drawn your attention to the fact that alternators are generally designed for high voltage. The reason is obvious. If we wish to carry the current, be it for lighting or power, to any distance, we must use a high voltage, in order to bring the section of our conducting wires or mains down to a size which makes the whole enterprise commercially possible. But to give our customers a current of some thousands of volts would be dangerous and inconvenient, for glow lamps require a current of about 100 volts when arranged in parallel, that is, in the way in which they are of most use to private consumers. The question, therefore, arises what to do with our high-pressure current when we have brought it to the place where its energy is required for lighting lamps. Obviously we must transform it; we must lower its voltage and increase its strength. Now there are two ways in which this may be done. We may use the current to work a motor, and use the power given out by the motor to drive a dynamo of 100 or 200 volts. The direct current can then be distributed to the lamps in the usual way, and we may even supplement the installation by secondary batteries, so as to be able to shut down our machinery during the hours of minimum demand. As far as I know, this system of transformation has only, up to now, been used in one installation, namely, at Cassel, in Germany. At the first glance it may seem complicated and costly, but it has many advantages, which will probably lead to its adoption in other towns.

The other system which is at present in general use is that of direct transformation by means of induction coils. Here we need no moving machinery, but simply a stationary apparatus consisting of a laminated iron core and two coils (*Fig.* 26). One of these consists of many turns of fine wire, and is technically termed the primary coil P, and the other of fewer turns of stouter wire called the secondary coil S. The high-pressure current is brought by the mains *w*, and the low-pressure current is supplied to the lamp L by the secondary mains W. You will observe that there is absolutely no connection between the two sets of mains, and this is a great

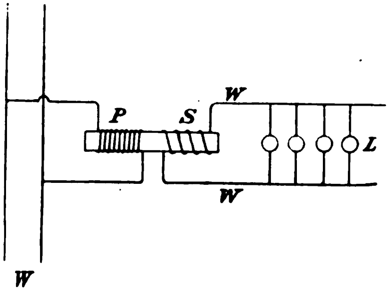


Fig. 26.

guarantee for the safety of the system. The action of this apparatus, which is technically known under the name of "transformer," will be clear to you from what I have said in the first lecture about the generation of an alternating electromotive force. The primary coil magnetizes the iron core in alternate directions, and at each reversal the lines of force cut through the wires of the secondary coil. The latter must, therefore, become the seat of an alternating electromotive force. If we denote by F the total induction, and call n the frequency, the maximum electromotive force generated in each turn of wire is

$$2\pi n\mathrm{F},$$

and the effective electromotive force is this value divided by the square root of 2. If the coil contains τ_2 turns, the total effective electromotive force in the large circuit will be

$$e_2 = 4{\cdot}45\ n\mathrm{F}\tau_2\,10^{-8} \text{ volts} \quad \ldots\ldots\ldots\ldots\ldots\ldots (19).$$

The changing induction affects, however, not only the secondary, but also the primary coil, and in the latter there will be developed an electromotive force which we compute by the same formula, only substituting for τ_2 the number of primary turns τ_1. You see, therefore, that the transforming ratio is given by the ratio of the turns of wire in each coil; but this is only approximate, since not the whole electromotive force generated in the secondary reaches its terminals. We must deduct the electromotive force used up in overcoming the ohmic resistance of the secondary coil. In like manner the electromotive force which opposes the current in the primary coil is a little smaller than the terminal electromotive force, because the ohmic resistance also opposes the primary current. The transforming ratio, therefore, varies with the load, but I may at once say that in good transformers the variation as determined by ohmic resistance is exceedingly small, generally about 2 per cent. There is, however, another cause of variation, namely, magnetic leakage, and a transformer made as shown in *Fig.* 26 would exhibit this phenomenon in a most objectionable degree. You see that the two coils meet in the middle of the core. Now the primary wants to magnetize the core in one direction and the secondary wants to magnetize it in the opposite direction. The result is that the two streams of induction come, so to speak, into collision about the middle of the core, and some of the lines which the primary coil tries to shoot through the secondary are squeezed out sidewise, and contribute nothing to the secondary electromotive force. You might, perhaps, think that this is of no moment, for we can make up for the loss of these lines by putting a few more turns of wire on the secondary. But if we did that we should get too much electromotive force at light loads. To see this clearly let us begin with no load on the secondary. Then there is no current in S and no collision. The lines created by the primary pass without opposition through the secondary, and F in Formula (19) has its full value. Now switch on some lamps and a secondary current will flow. We shall have some collision of lines and F in the formula, and therefore the electromotive force will become smaller. The more lamps you switch on, the more current flows through both coils, and the more violent becomes the collision, and, therefore, the number of lines lost. In a word, we generate more lines than we can utilize. The obvious remedy for this defect is to place the coils relatively to each other into such a position that the lines generated by the primary cannot evade passing through the secondary. For instance, we can wind one coil over the other, or we can split up the coils into short sections and place them alternately over the core. Even with these precautions there is some magnetic leakage, but this does not as a rule lower the voltage by more than 1 or 2 per cent. Thus in a good transformer we may expect to get, with constant primary voltage, a terminal pressure varying between 102 and 99 or 98 volts, when the load is increased from zero to full out-put. These figures refer to small transformers of 50 or 100 lamps. With large transformers it is quite possible to limit the total voltage drop to something under 2 per cent. The transformer shown in *Fig.* 26 is defective in other ways besides its great voltage drop. The lines passing through the core have to come back through air, and the great magnetic resistance of their path through air requires a strong magnetizing current, or, in other words, the primary current will be considerably greater than in a transformer, in which the return path is made more easy. One way of doing this is to increase the surface of the core ends, and this Mr. Swinburne has done in his "Hedgehog" transformer. The core consists of iron wires which at the ends are curved outwards. Thus part of the return path is through iron. Another method, and this is generally used, is to make the whole return path of iron. We may, for instance, employ a closed iron frame (J, *Fig.* 27), and wind the primary and secondary coils C over each other on two opposite sides of this frame. The iron frame or core is composed of thin plates, more or less insulated from each other to avoid eddy currents. This type of transformer is called a "core transformer." Or we may employ only one coil and surround it by a double frame, as in *Fig.* 28, a kind of iron shell, and this construction is called a "shell transformer." Both figures have been drawn to represent transformers of equal out-put. The depth of the core is supposed to be equal, and its width in *Fig.* 28 is twice as great as in *Fig.* 27, to make up for there being only one coil. At the first glance it is difficult to say which is the better transformer, though practically the balance of advantages seem to lie with the shell type, which is most in favour with the makers of this kind of apparatus. If we enquire what it is we must aim at in the design of a good transformer, we find that the length of wire should be small in order to reduce ohmic resistance and cost, that there should be as little iron as possible, and that the magnetic circuit should be short. Now these are contradictory conditions. To reduce the length of wire we must work with a high total induction, so that a small number of

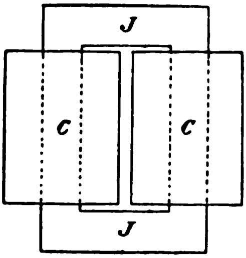


Fig. 27.

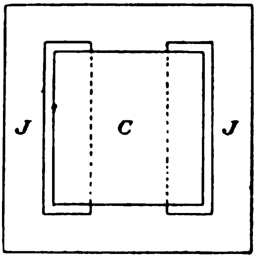


Fig. 28.

turns should give us the required electromotive force. But a large induction means either a great loss by hysteresis or a stout core, and a stout core means that the length of each turn of wire is great. It further means a longer magnetic circuit and a greater weight of iron, which again increases the loss by hysteresis. You see here again the successful design must be a compromise, but a compromise in which a preponderance of weight is given to hysteresis. We must remember that a transformer is continuously at work whether we take current from it or not, and hence the hysteresis loss goes on day and night. Even an extra loss of ½ per cent. by hysteresis will, therefore, be felt in the all-day efficiency of the apparatus, and is, therefore, more serious from an economical point of view than a loss of several per cent. by copper resistance, because transformers, when used for lighting, only work a very short time daily under full load. On the other hand, we must not allow an excessive copper loss, as this would disqualify the transformer on account of too great a voltage drop. We are thus hemmed in on all sides by conflicting conditions, and the design of a good transformer is by no means so easy a task as might appear at the first glance. As a starting point, we may take it that the magnetic and the electric circuit should be as short as possible, and this condition will be best fulfilled by a circular or square shape, or as near an approach to such a shape as possible. In fact, if you examine the successful transformers in the market, you will find that this condition is fulfilled in either one or the other circuit, but not in both. I have not succeeded in establishing, and I can, therefore, not give you any, hard-and-fast rules for the construction of transformers, but in order to enable you to see what enormous influence slight alterations in the proportions have on the weight and cost of the apparatus, I have prepared for your proceedings some 27 different designs, all for a 100-light transformer. Four of these designs are given full size in *Plates* I. and II. In all these the copper loss is 2 per cent. The hysteresis loss is given in each case. You can see at a glance what a great difference there is in the amount of copper required, and how by a skilful choice of the proportions the cost of the apparatus can be reduced without lowering its efficiency.

Before concluding this part of my subject, I wish to draw your attention to the relation existing between the linear dimensions of a transformer and its out-put and hysteresis loss. Imagine that after designing a score or so of transformers you have at last arrived at a type with which you feel satisfied; but suppose it is not the size you want. How will you alter its linear dimensions? Let us try what we shall get if we make everything twice as big, including the size of the wire. We retain the induction per square centimètre at the 4,000 or 5,000, which we found will give us a loss of say 1½ per cent. by hysteresis. We also retain the number of turns in both coils. The total induction is then four times as great, and the electromotive force also four times as great. The resistance of the coils has been reduced to one-half its former value (area of wire four times as great, and length twice as great). If we are satisfied to have the same copper loss, we can now allow a current which will give us four times the previous voltage loss, but as the resistance has been halved the current will be eight times its former value. Thus the current is eight times and the volts are four times as great as before; the out-put will, therefore, be 32 times as great. But 32 is two to the fifth power, and hence we see that the out-put of a transformer varies as the fifth power of its linear dimensions. The weight, cost, and hysteresis loss, on the other hand, all vary, as the cube of the linear dimensions and the weight per kilowatt of out-put varies inversely as the square of the linear dimensions. Or, in figures, if 40lbs. of copper and iron were required for each kilowatt produced by the small transformer, only 10lbs. per kilowatt will be required in the larger; and if the small transformer wasted 2 per cent. of its power in hysteresis, the large transformer will only waste ½ per cent. Let the large transformer have x times the out-put of the small one, then linear dimensions must be proportional to $x^{\frac{1}{5}}$. Weight and cost per kilowatt and percentage loss by hysteresis will be proportional to $x^{\frac{2}{5}}$. This calculation neglects, however, the working temperature which, for obvious reasons, must not exceed a certain limit. In practice it is found that for every watt lost by hysteresis and ohmic resistance a cooling surface of from three to four square inches must be provided. As the larger transformer has, relatively to its out-put, a smaller external surface than the smaller transformer, it is not possible to take full advantage of the loss between linear dimensions and out-put here given.

VII.—CENTRAL STATIONS AND DISTRIBUTION OF ALTERNATING CURRENTS.

The principal reason for the use of alternating currents in connection with the supply of light from a central station to private customers is that in consequence of the high pressure which can safely be used we are able to take on customers, whether near or far, and thus carry on the business of light purveyors on a larger scale, and presumably with a greater profit, than if we were restricted to those customers only who live near the station. In a general sense this argument is perfectly sound, but it would be a mistake to apply it indiscriminately and say that in all cases the supply by means of alternating currents is preferable to that by continuous currents. Whether an engineer has to design works himself, or merely to inspect and approve works carried out by others, it will always be his first concern to see that the works shall be a commercial success. We cannot build central stations or any other works without the aid of the financier, and the financier cares very little for any technical perfection; all he cares for is that the work should pay, and unless the engineer can give him that assurance he will not co-operate. Hence it is the business of the engineer not only to design his works so as to be technically a success, but also commercially.

In considering the relative merits of the two systems, we must take into account a variety of local circumstances, some of which not only are beyond the reach of mathematical representation, that is, representation by concrete figures which we can use in our calculation, but may even be but vaguely known at the time the station is being designed. For instance, the number of lamps which will be required in any given district, the daily lighting time of each lamp, and the distribution of lamps between the different classes of houses in the district, are matters which we cannot foretell with absolute certainty. We can but make a guess based on previous experience. Another matter of some importance, but about which it is extremely difficult to form an estimate beforehand, is the danger of being served with an injunction for noise or vibration by some of the kind neighbours, who are always on the look-out how to make a little money out of the difficulty of others. This danger is evidently greater in the direct current system, because with it we have not a very wide choice as to the position of our station, but must place it fairly near to and preferably into the centre of the district to be lighted. With the alternating current system we can afford to go farther afield with our station, into a neighbourhood the inhabitants of which are not so particular as to noise and vibration. Then there is the question of the total extent of the district to be lighted, the possibility of working by water power, or if not, the cost of coal and water, the quality of the latter, the possibility of obtaining condensing water, and many other points which have to be considered.

If we have to do with a compact and densely lighted district, where most of the lamps can be placed within a few hundred yards of the station (or at any rate within a radius of about 1,000 yards), then the direct current system is generally the best. One of its greatest advantages lies in the fact that we can supplement the dynamos by storage batteries, and use the latter during the hours of minimum demand. For economical reasons we are obliged to use compound engines, but, as you know, a compound engine, except when condensing, does not work with economy when lightly loaded, and it is, therefore, advantageous to shut down the engines altogether in the early hours of the morning and during the daytime, putting the batteries on for the supply of the few lamps required. In this respect the direct current has a distinct advantage, but this advantage becomes less and less felt as the total power of the station is increased, because in a large station the number of lamps, even in the daytime, will be large enough to fairly load a small engine, and if we can obtain condensing water the engine, even if only partly loaded, will work with fair economy.

A point at present in favour of direct currents is the ease with which they can be used for motive power, but there is every prospect that ere long alternate current motors will become a practical success. At any rate, the use of motors on town circuits has with us not yet become so popular that we need attach any great weight to this point. The principal advantage of the alternating current system is that we can use small mains, and yet keep the pressure throughout the district very nearly constant. With continuous currents not only do we require more copper in the mains and feeders, but where the feeders are long, the loss of pressure in them amounts sometimes to as much as 20 per cent. of the total or station voltage, and in such cases some complicated arrangements are required for the regulation of the voltage, so as to keep the pressure at the feeding centres at least approximately constant.

There are two ways in which we can use transformers. We can bring the high pressure mains right into the house of each customer and give him his own little transformer, or we can place large transformers at certain sub-stations, and lay through the streets a second system of low-pressure mains, with house connections, in the same way as if the supply were by direct currents, only that in this case the low-pressure mains need not be so large, since we can put down as many sub-stations as we please, and thus reduce the distance to the lamps to any desired limit. The system of a separate transformer for each customer has hitherto been most used, but it is not the best. It is true that by it we save the cost of the secondary mains and the cost of the sub-stations, items which a company in its early pioneering days, when customers are few and far between, could not easily afford. On the other hand, the objections to the use of separate transformers are great, and as time goes on, that is to say, as the use of the electric light extends, these objections acquire additional weight. In the first place, there is some danger in having a high-pressure apparatus in one's house. You may put your transformer into the cellar in a fireproof case and lock it up, but when you have thousands of transformers in as many houses, the chances are that in one or two cases the locking-up may be forgotten, and some inquisitive person may touch a terminal. A further objection lies in this: that a number of small transformers cost more money and waste more energy than one large transformer. Let us take for example twenty houses, each wired for fifty lamps. Each house must get its 50-light transformer. The whole of the fifty lamps will not be lighted simultaneously every day. Probably not more than half-a-dozen times in the year will each transformer be worked at its full out-put, and there is the hysteresis loss going on in it day and night. This loss means waste of power and development of heat; indeed, I have heard of one case in which the heat given out by a transformer placed in a wine cellar was sufficient to keep the cellar at a nice even temperature all the year round. General experience tells us that scarcely more than ½, or at most 60 per cent., of the lamps wired in a district are ever alight simultaneously. The maximum joint demand for current of our twenty houses will, therefore, never exceed 600 lamps, and we can substitute for the twenty separate transformers of fifty lights each, one single transformer of 600 lights. From what I have said before about the influence of size on the cost of transformers, you will see that the single large transformer will cost scarcely more than a third the money required for the twenty small ones, and that even if we put down two large transformers so as to keep one in reserve, we shall do it for little more than half the money. Similarly, the loss of power by hysteresis will be reduced to one-quarter, and this is a very important consideration. Take for instance a station designed for 20,000 lamps, of which 12,000 will be alight simultaneously during the two or three hours of maximum demand. The average lighting time of each lamp fixed is in London about 500 hours per annum. If we allow with small transformers a loss of 2 per cent. by hysteresis, the power continuously absorbed by all the transformers connected to the central station will be equivalent to that required by 400 lamps. We are wasting, therefore, day and night, current which could feed 400 lamps. In a year we waste not less than 3,500,000 lamp hours, whereas our total income from the 20,000 lamps is only 10,000,000 lamp hours. This means that even if there were no other sources of loss we would have to send out energy from our station representing 13,500,000 lamp hours, but we could only get paid for 10,000,000 lamp hours. This is only 74 per cent. efficiency. Now suppose we use sub-stations and large transformers, the hysteresis loss will fall to ½ per cent., and the efficiency will rise to over 90 per cent. We can further improve the efficiency by putting down at the sub-station not one transformer only, but two or more of different size, and make arrangements for the insertion or withdrawal of tranformers from the two circuits (the high and low-pressure circuits), in accordance with the demand for current, so that during the hours of light load the hysteresis loss will only take place in the smallest transformer of the group. Mr. Ferranti, Mr. Gordon, and myself have, independently of each other, devised an apparatus which switches in and takes out the transformers automatically.

The employment of large transformers at sub-stations has this further advantage: that the total length of high-pressure mains is thereby considerably reduced, and that there are no branch connections on these mains. We are thus able to get higher insulation. You know that an insulation of many hundreds of megohms per mile can be easily attained in a continuous cable, but after the cable has been laid, and branch connections have been made, the insulation is much lower, the reason being that at every joint the insulation has first to be stripped, and then made good again. Now it is one thing to put on the insulation in the factory, where every precaution can easily be taken to ensure perfect work, and it is quite another thing to do the same kind of work at the bottom of a trench or pit in the street. No matter how careful we are, the insulation put on under these circumstances can never be so good as that put on by the covering machines in the cable factory. For this reason a system of simple mains radiating from the central stations to the sub-stations must show a higher insulation than a complicated network of mains covering the whole district.

I have given you here the main reasons for the adoption of transforming sub-stations in connection with alternate current distribution, but, in applying them to each given case, you must not forget to take into account the commercial element. A system of working may be scientifically the best, and yet not the best financially. Thus the system generally applied at the present time in London in alternating current stations is that of a separate transformer for every customer, not because it is theoretically the best, but simply because it is commercially the only feasible system. I have, however, no doubt that as the use of the light becomes more general, the various companies will find it advantageous to change to the system of sub-stations. I have hitherto not said anything as to the comparative cost of continuous and alternate current stations, and it is indeed very difficult to state it in any definite way. The cost of boilers, engines, and accessory apparatus will be about the same in both systems. The alternators will be a little cheaper than the dynamos, and there will also be the cost of the battery against the continuous current system. On the other hand, we have to remember that the whole engine power may be slightly less, since during the hours of heavy lighting the battery assists the engines. Taking, then, one thing with another, there will not be any very great difference in the cost of the plant at the central station on the two systems. The difference is mostly outside. If the district is large, the extra cost of the heavy feeders and mains will, with continuous currents, be much greater than that of the high-pressure feeders and low-pressure mains if alternating currents are used, and the margin left will be more than sufficient to pay for the transformers at the sub-stations. If the district be small, and the lighting compact, then the balance will be the other way. The cost of mains will be about the same in the two systems, but we shall not be able to save enough on the cost of the feeders to pay for the cost of the transformers. Each case must, in fact, be judged on its own merits, and what I have said here about comparative cost is merely intended as a guide in forming such a judgment.

VIII.—EXAMPLES OF CENTRAL STATIONS.

As examples to illustrate this lecture, I choose three types of central stations, distinguished from each other mainly by the character of the motive power employed. In the first type the motive power is steam, in the second it is water power, and in the third it is electricity.

1. *The Sardinia Street Station of the Metropolitan Electric Supply Company.*—The boilers are of the Babcock Wilcox type, and placed

on the ground-floor. The battery of boilers is parallel to the two rows of engines in the adjoining room, also on the ground-floor, but at a slightly higher level. This arrangement of boilers and engines has the great advantage of reducing the length of steam-piping, and minimizing the inconvenience resulting from a failure of any particular length of steam-pipe. The steam-pipe forms what is technically termed a ring main, and valves are inserted at suitable points, so that any length can be cut out without disturbing the supply of steam through the rest of the piping. Adjoining the boiler-room, and connected with it by a tram line, is a vast underground coal store; a very admirable arrangement, especially in a station situated, as is that of Sardinia Street, in a district where coals can only be delivered by cart, and where, consequently, the delivery may, in times of heavy frost or fog, be interrupted for some days or weeks. The engines are of the compound high speed Westinghouse type, and drive by belt Westinghouse alternators placed on an upper floor. Alongside one wall of the machine-room is placed the switchboard, by means of which any desired combination between the alternators and external circuits can be quickly made. During the hours of light load all the circuits are put on to one or two machines, but as the load increases other machines are started, and some of the circuits are transferred to them. The machines are not worked in parallel. As regards the mains, I must mention an ingenious arrangement due to Mr. Bailey, the engineer to the company. In order to avoid the difficulties connected with the insulation of joints, when these are made in the streets Mr. Bailey makes, as far as possible, all connections of the high-pressure mains by terminal blocks on the customer's premises. Under this arrangement the insulation is only stripped at the ends which enter the terminals, and which themselves can be perfectly insulated. It is true that under this arrangement the total length of cable required is slightly increased, namely, by the length of the bight taken into each house; but this is only a small percentage of the straight run of main. Further, we have the advantage that each house is, so to speak, served by duplicate mains, namely, one on either side, and that, therefore, the house need not be cut off even if one length of main in the street should for any reason have to be disconnected. We have here, in fact, the electrical equivalent of the ring main between the engines and boilers.

2. *The Lynton Station.*—This is worked by water power from the River Lyn on a fall of 96 feet, the water being supplied to the

turbine through a 30-inch pipe. Owing to the high fall the speed is sufficient for driving the alternators, which are Mordey machines directly coupled one on either side of the turbine. Each alternator is capable of developing 37·5 kilowatts. The speed is regulated by a slide valve in the main water supply pipe worked by a hand-wheel. The mains are lead-covered Callender bitumen cables laid underground.

3. *The Keswick Station.*—This is also worked by water power obtained from the River Greta, but since the fall is only 20 feet, the alternators are driven by belt from the turbine shaft. The plant comprises two 30-kilowatt Kapp alternators, the necessary exciting machine, switchboards and instruments, and a boiler and Westinghouse engine to serve as an auxiliary source of power in case of drought or frost. The mains are insulated cables placed overhead on oil insulators, but for a certain distance they had to be taken underground, and then a Brookes' pipe line is used.

4. *The Cassel Station.*—This is an example of a central station where the motive power is electricity. There are two stations in the town in which dynamos are driven by Kapp alternators working as motors. The alternating current is supplied from a water-power station four miles distant. *Fig.* 29 shows the arrangement diagrammatically. At the power station a turbine drives two Kapp alternators, each designed to give 30 ampères at 2 200 volts. The machines

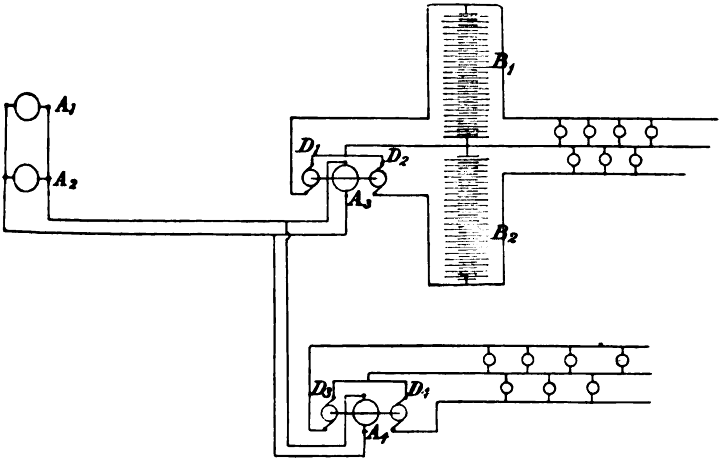


Fig. 29.

are coupled in parallel, and the current is taken by a concentric lead-covered cable to Cassel, where the cable splits into two branches, each leading to a lighting station. At each of these lighting stations there is a 60-kilowatt alternator coupled direct to two 30-kilowatt dynamos wound for 110 volts, one dynamo on each side of the alternator. The dynamos are arranged on the three-wire system, and work on to a three-wire network common to both stations. At one of the stations there is also a battery of storage cells, from which the town is supplied during the hours of minimum demand. Towards evening, when it is necessary to supplement the batteries by dynamo power, or when it is desired to re-charge the batteries, the dynamos are switched on to the network, and receive current from it. This sets them in motion, and, working for the time being as motors, they run up the alternators to synchronizing speed. The alternating current is then switched on, and the action between the machine is reversed, the alternators acting as motors, and driving now the dynamos. The two stations supply at present current for 2,600 16-candle-power lamps burning simultaneously, or 3,500 lamps wired, but provision has been made to extend the plant, so as to eventually supply 12,000 lamps wired.

IX.—Parallel Coupling of Alternators.

I have already pointed out that for economical reasons it is advisable to work the engines at a station as nearly as possible at their full load, and you will easily see that this condition can most easily be fulfilled if the alternators can be worked in parallel. For were it only possible to work each machine quite independently of the other machines, we should be obliged to keep a larger number of machines working on small loads, and as the hours of light load greatly exceed those of full load, the engines would be used under very uneconomical conditions. But if we can couple the alternators parallel, then we can put on and take off machines exactly in accordance with the demand for current, and have our engines fairly well loaded at all times. Some years ago it was believed that alternators had to be designed specially for working in parallel, and certain makers claimed this quality of their machines as something specially in their favour. If parallel running succeeded it was put down to the credit of the particular type of alternator; if it failed, the design of the alternator was considered faulty. It is only recently that we have come to recognize that the real difficulty of parallel running is not in the alternator at all, but in the engine. Any alternators, when driven by turbines which have an absolutely constant angular speed, will run parallel perfectly, but if you drive the machines from slow speed steam engines by means of belts or ropes, any irregularity in the angular speed of the engines is magnified by reason of the multiplication of speed, and the machine becomes alternately a generator and a motor, the transition from one state to the other being accompanied by heavy mechanical and electrical strains, which render anything like smooth working impossible. The condition of successful parallel working is, therefore, a direct-coupled engine having a very even angular speed. This is a point of great practical importance, and it is intimately connected with the general question of alternators used as motors, since when the engine fails to keep up its even angular speed the alternator steps in and compels it to do so; it acts, in fact, for a moment, as a motor, and controls the engine. This brings me to the subject of

X.—ALTERNATING CURRENT MOTORS.

When investigating the transmission of power by alternating currents, we may consider the circuit as consisting of three parts: a line having a definite resistance; an alternator working as generator at one end; and another alternator working as motor at the other end. Such a conception would be the most obvious, but it is not the best, because we are thereby compelled to investigate simultaneously the behaviour of two machines. To simplify the treatment I shall assume the following case:—Given a pair of terminals, between which by some means we maintain a constant alternating electromotive force at constant frequency, and the source from which this electromotive force is supplied shall be so abundant that we may take any amount of energy from the terminals, or put any amount of energy into them, without altering in any way either the pressure or the frequency. Such a pair of terminals would, for instance, be the omnibus bars at a central station, if from them we supply a small motor. Suppose the motor run up to synchronizing speed, and then switched on to the omnibus bars. We now want to know the relation between the mechanical power obtained, the current through the armature, and the strength of field. This apparently complex problem can be solved graphically by means of a clock diagram in a very simple manner.

It is self-evident that we can only obtain power from the motor if it runs at such a speed that the frequency of the electromotive force developed in its armature coils is exactly the same as that of the current which drives it, and that the electromotive force of the motor must be opposed to the current.

We must, therefore, at first employ some external source of power to run the motor up to the proper speed before switching on. But how are we to know when the proper speed has been reached? No tachometer or speed counter can give us this information with sufficient accuracy, especially since there may be slight variations in the frequency of the supply current. If we wish to put two alternators in parallel, we also must know exactly when their phase and frequency coincide, and for this purpose we use an instrument called a "synchronizer." It consists mainly of two small transformers (*Fig.* 30), the primaries of which are connected to the terminals of the two alternators which are to be coupled parallel. Two of these terminals may, of course, be permanently connected, as shown in the figure, but the other two must only be connected by the switch S when the machines are in step. The secondaries of the two transformers are connected as shown, and into this circuit are placed some incandescent lamps. By following out the connections you will easily see that if the two machines are in opposite phase, that is, in a condition when you must not couple them, there is no electromotive force on the lamps, but that when the machines are in the same phase, or, as we call it, "in step," then the lamps get the full electromotive force of the two transformers coupled in series. We thus know that when the lamps are dark the machines are out of step, and when they light up the machines are in step. But complete darkness or complete brightness can only occur when the frequencies are absolutely the same. Generally, the frequencies

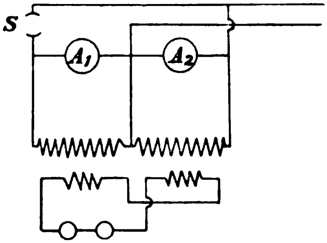


Fig. 30.

will be different, and the lamps will flicker. Thus, suppose one machine is running at its normal speed, and the other is being started up, at first there will be very rapid flickering in the lamps. As the speed of the second machine increases, the flickering becomes less rapid, and by degrees, namely, as the speed approaches that which is required for synchronism, there appear regular beats in the light of the lamps, which get longer and longer. You watch your opportunity, and throw the switch on in the middle of a beat when the lamps are alight. The machines are then so nearly in the right step that the first rush of current pulls them dead into step, and they remain, as it were, interlocked in that condition. The electrical coupling is, in fact, comparable to a kind of interlocking, which is as secure as if the two armature spindles were connected by spur gearing. To test the reliability of this electrical interlocking, I have run a 60 and a 10-kilowatt alternator in parallel, supplying the power from two independent sources. I have then cut off the power from the small machine. It ran on exactly as before. Next, I put a load on the small machine, and increased it to 25 horse-power; still the machine ran on. I left the load on for some hours, and then suddenly withdrew, and again put on a large portion of the load, but the machine kept in step. There is, of course, for every machine a certain load at which it will be torn out of step, just the same as there is for every spur wheel a load, which will strip its teeth. In the machine with which I experimented it should be possible to break down the synchronism with a load of about 30 horse-power, that is, twice the normal load, but I was not able to determine the breaking off load experimentally, because the belt by which power was taken from the motor begun to slip at 25 horse-power. The machines with which I experimented were of my own type, but, as I have said before, there is no particular virtue in the design. Any modern alternator with a smooth armature core, and having a fair efficiency, will behave in exactly the same manner.

Having now given you some practical results of parallel running and transmission of power, I must briefly explain the theory of it. *Fig.* 31 represents the condition of a machine supplying current to a non-inductive resistance. OB is the electromotive force which it would at its then excitation give on open circuit, AB is the electromotive force required to overcome its self-induction with the current it actually gives, AR is the electromotive force required to overcome the armature resistance, and RO is the electromotive force available for the external circuit. If in a central station we have already a number of machines running in parallel, OR would be the electromotive force on the omnibus bars, and if we wish to switch in a new machine we would, in order to have it in the same condition as the others, excite it to such a degree that on open circuit it will give the electromotive force OB. We run the machine up to the right frequency and switch it on. For the sake of the present investigation, I will assume the new machine can be mechanically geared with one of the other machines in such a way that its electromotive force shall lag or lead in comparison with the omnibus electromotive force by any desired angle. Thus in *Fig.* 32 OR represents the omnibus electromotive force, and OB the machine electromotive force, the angle between the two being ensured constant by the mechanical gearing. By drawing the parallelogram ORCB we find the resultant electromotive force in the new machine OC, and this can be regarded also as the resultant of the electromotive force of self-induction CD, and that lost in armature resistance OD. If we regard the coefficient of self-induction constant, then the angle ϕ, which OD makes with OC, is the same as that which in *Fig.* 31 RA makes with RB, and the direction of the line OD in *Fig.* 32 is at once defined. The point D is found by dropping a perpendicular from C on this line. Since OD represents the electromotive force used up in resistance, and since we know the resistance, it is easy to calculate the current. We know then the direction and magnitude of the current as well as of the electromotive force, and we can find the work done by the machine. For this purpose we multiply the current with the electromotive force, and with the cosine of the angle enclosed between the two lines. The work thus found, we mark off on the line OB or its prolongation. Now let us shift our mechanical gearing and find in the same manner the current and work for a different angle between the omnibus

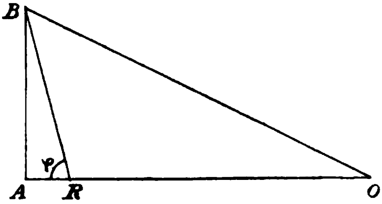


Fig. 31.

and machine electromotive force, and repeating the construction for various phase angles, we obtain the curves on *Fig.* 33, which show current and work as functions of the phase angle. The outer curve on the left represents the work given out by the machine when its phase angle lags from O to about 180°, the inner curve represents the work absorbed by the machine (when working as motor) when its phase angle lags from 180 to 360°. The curves on the right represent similarly the work given to or taken from the omnibus bars. You will notice that about half of the curves are dotted. The dotted parts refer to an unstable condition of working, and the diagram shows at a glance why it is impossible to run two alternators in series if they are independently driven, that is, not mechanically geared together, as I have assumed to be the case for the purpose of explaining how this diagram is obtained. You will also

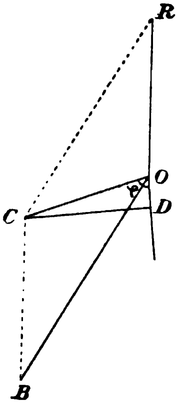


Fig. 32.

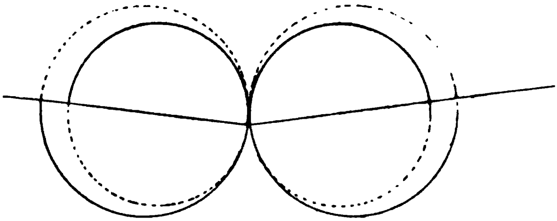


Fig. 33.

see that a moderate difference in the phase angle is sufficient to transform the machine from a strong generator into a strong motor. The difference of position in the two cases is about 90°, but you must remember that the diagram represents a two-pole machine. In reality the machines are made with many more poles, and the angle is much smaller. For instance, if there were eighteen poles the angle would only be about 10°, and this explains why it is essential for parallel working, and also for power transmission, to employ engines which will impart to the machines an almost absolutely constant angular velocity.

XI.—SELF-STARTING MOTORS.

From what I have here said you will conclude that there is no difficulty in transmitting power by a single alternating current, but that the motor is not self-starting. The system is thus encumbered by the necessity of providing a separate starting machine and some storage of power to set this in motion. The most convenient way is to use the exciter for this purpose, and drive it by a storage battery. When the alternator is working as a motor it drives its own exciter, and the latter may at the same time be used to charge the battery up again ready for the next start. The complication and cost of this arrangement are not very serious objections when we have to deal with large powers, but for the distribution of small parcels of power the necessity of providing a separate exciter and a storage battery, in addition to the motor proper, is a fatal objection, and various attempts have been made to design a self-starting alternate current motor. One of these, and I may at once say, the most successful one, is due to Mr. Zipernowsky, whose firm (Ganz & Co., of Budapest) showed at the Frankfurt Exhibition several of these machines at work. In the limited time at my disposal I cannot attempt to give you a detailed description of these, nor enter into the many refinements of construction which have been found necessary in developing the machine practically. I must content myself to give you the main principle of it. In *Fig.* 34 M is a laminated magnet and A an armature wound with a single coil, the ends of which are brought to a two-part commutator. It is, in fact, the well-known Siemens' shuttle armature, also employed in the small Griscome motor, and the apparatus, as here shown, is nothing else than a very simple form of continuous current motor, which is self-starting from almost any position. The only position when the motor will not start is when the armature is placed so that the brush on each side touches both commutator segments at once. To start the motor from this position, it is, of course, necessary to slightly shift the brushes to one side or the other of the dead centre. From what I have said in the first lecture, you will easily see that this kind of motor will also start and work with an alternating current, but its power will at first be very slight. Observe now what happens when the alternating current is switched on whilst there is no load on the motor. It will start and gather speed as all series wound motors do. If the current were continuous, the motor would very soon begin to race, but with an alternating current this cannot happen, because in trying to get up a racing pace the armature must pass through that speed which corresponds to the frequency of the supply current. At the moment when this happens, the reversal of current produced by the commutator coincides exactly with the reversal of the supply current, and the result is that the current flowing through the armature does not any more change its direction. The armature has virtually been transformed into a field-magnet, excited by a continuous current, and what was at starting the field-magnet has now become the armature of an ordinary alternator. The moment when the machine jumps into step can be easily noticed by the behaviour of the brushes. At starting there is violent sparking and a peculiar noise. As the machine gathers speed the sparking gets less, and suddenly there is a kind of jerk, after which both noise and sparking cease and the load may be put on. The motor, when once in step, will even stand a considerable overload.

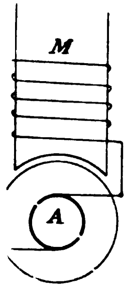


Fig. 34.

XII.—MULTIPHASE CURRENTS.

The motor I have just described will start itself, but it will not start with a load. The sparking is also an objection which renders the machine useless for flour mills, cotton mills, and any works where an explosion may be caused by sparks. We can, therefore, not regard this motor as the final solution of the problem of transmitting power by alternating currents, but must look for the solution in quite another direction. This direction has been first indicated by Professor Galileo Ferraris, of Turin, some six years ago. Quite independent of Ferraris, the same discovery was also made by Nicola Tesla, of New York; and since the practical importance of the discovery has been recognized, quite a host of original discoverers have come forward, each claiming to be the first. With these various claims we need not concern ourselves at present. I will merely describe the apparatus used by Ferraris. He employed two vertical coils AB (*Fig.* 35) set at right angles to each other, and a copper cylinder C suspended between them. Two alternating currents of the same frequency, but with a phase difference of 90 degrees, were sent through the two circuits, and the copper cylinder was thereby set in rotation. The explanation is as follows:—Each coil taken by itself produces an oscillating magnetic field, the lines of which are at right angles to the face of the coil. The two coils together produce a resultant field which revolves round the vertical axis of the apparatus. The surface of the copper cylinder is therefore being continuously cut by lines of force as they sweep round; currents are induced in the copper which by Lenz's law are in such directions as to resist motion; and since the cylinder is freely

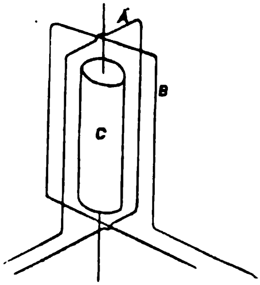


Fig. 35.

suspended, its endeavour to resist the motion of the field results in its being set in motion itself.

Experiment Lantern and Model.—In translating this laboratory experiment into practical work we must, of course, make many alterations and improvements. We must, for one thing, employ iron to get a more compact and a stronger apparatus. We must also sub-divide the two coils in order to get a machine which will run at a moderate speed, and finally we must substitute for the plain copper cylinder an armature properly wound. A machine designed on these lines will be, generally speaking, a great improvement on the original apparatus, but in one respect it will not be so good. In *Fig.* 35 the coils are at right angles, and the currents are at right angles. As you have seen by the model, the effect of this combination is an absolutely constant magnetic field revolving round the axis with constant speed. But if we split up the two coils into a number of sections, and wind these alternately side by side on a cylindrical core, as we wind a Gramme armature, one of our conditions, namely, that of the right angular position of the two coils, has been lost, for the coils are now very nearly in line with each other all the way round, and the result is that the field is not any more absolutely constant. I can show you this by means of the model. By setting the cranks at the wrong angle you see immediately that the vector of the resultant field is no longer the radius of a circle, but of a curve resembling an ellipse. To find the variation in the strength of the resultant field we need only draw the two current curves and add up their ordinates as I showed you in the first lecture. If you do this you will find that the maximum strength of the field is about 40 per cent. greater than its minimum value. The field is now not only a rotating one, but it also pulsates. The rotation is what we want, but the pulsation is objectionable, as, in consequence of it, the whole machine acts partly as a huge choking coil, and the power obtainable from it is thus reduced. It is the great merit of Herr von Dobrowolsky to have been the first to clearly recognize this defect in machines based on the Tesla-Ferraris motor. The evil once understood, a remedy was soon found. Dobrowolsky adopted three currents instead of two, and thus reduced the pulsation of the field at once to something like 14 per cent.; but even this was not quite satisfactory. He went, therefore, a step further and re-arranged the winding of the field in such a way as to produce the effect of six distinct currents, though still only using three wires in the line of transmission. A reference to *Fig.* 36 will make this clear. Let *abc* represent the three coils of a two-pole drum armature, such, for instance, as the armature of a Thomson-Houston machine, but instead of joining the coils to a common centre and to a three-part commutator, as is done in this type of machine, let them be joined as shown, and let the points of junction ABC be three contact rings by which the currents are received from the line. According to Kirchoff's law, the algebraical sum of the three currents ABC must at all times be zero, for if this were not the case there would be an accumulation of electricity in the machine which is obviously impossible. Any of the currents may, therefore, be regarded as the resultant of the other two currents. Here we have a simple three-phase winding and a rotating field, the pulsations of which are about 14 per cent. of its minimum strength. Now to reduce these pulsations, Dobrowolsky adopts the following expedient. Instead of bringing the junction between *b* and *c* direct to the contact ring A, he attaches to the junction a stouter wire and winds this round the armature in a coil placed midway between *b* and *c*. Similarly B is wound so as to split up the phase difference between *a* and *c*, and C is wound in between *a* and *b*. We have now six coils on the armature, but only half the former phase difference between neighbouring coils. *Fig.* 37 shows a two-pole armature so wound, and in this way the pulsation is reduced to about 4 per cent. Were the winding not split up in the manner shown, the tendency to produce fluctuations in the strength of the resultant field would cause currents to circulate in the induced part of the winding, and these currents would prevent, to a certain extent, the fluctuations. But as they must necessarily circulate in coils which at the moment cannot contribute anything to the torque by reason of their position at right angles to the resultant field, these currents represent simply so much waste of power by ohmic resistance. Hence Dobrowolsky's

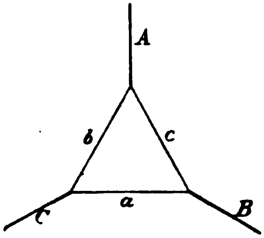


Fig. 36.

method of splitting up the winding, although not indispensable, is a useful device for increasing the out-put obtainable from a given mass of iron and copper, and for increasing the efficiency. I have called the part of the motor represented by *Fig.* 37 an armature, but this was merely to point out the analogy with a Thomson-Houston armature. It would be more correct to call this part, which receives the currents from the line, the field, because its function is to produce the revolving field. The armature of the machine is a hollow cylinder of laminated iron, built up of thin plates in the usual way, and provided with a winding which is closed on itself. To understand the principle of this winding, imagine a Gramme ring, the winding of which is altered in the following way. Instead of joining the inner end of each coil with the outer end of the next coil, so as to produce a spiral winding, let the two ends of each coil, be joined together. You will then have covered the Gramme core with a number of distinct coils, each closed on itself. Now put a field magnet into the inner space of the armature and revolve this magnet. The poles sweeping past the closed coils of the armature will create in them very powerful currents, and the mechanical reaction of these currents on the poles will require the application of a considerable twisting couple or torque to keep up even a moderate speed. You can test this for yourselves very easily by means of any continuous current dynamo. Excite its field separately, and short-circuit the brushes by a thick wire. If you then turn the armature by hand you will find that even exerting considerable force it will only creep round slowly, and you will thus

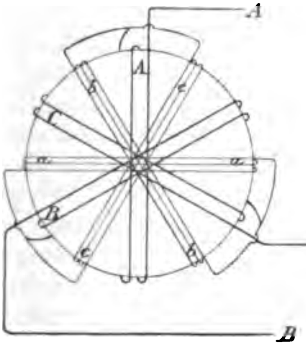


Fig. 37.

realize how a great torque may be developed by a small angular speed of the armature in relation to the field magnet. This is an important fact, and helps us to understand two things in connection with rotary field motors. The first is that the speed of such motors does not vary much when the load varies, since small variations of the relative speed between field and armature produce large variations in the torque; and the second is that the torque at starting is very large, the reason being that at starting the relative speed between armature and field is a maximum. It is, however, necessary to observe here that to get this large torque, resistance must be inserted in the armature circuit, for were this not done, the current in the armature coils would be so strong as to demagnetize the revolving field, thus again reducing the torque. We may now go back to *Fig.* 37, and see how this works out in practice. You have seen how a three-phase current passing through the winding produces a sensibly constant field, which revolves round the centre with a speed corresponding to the frequency. The armature surrounds the part shown in *Fig.* 37, but is omitted from the diagram. The lines of the field, in sweeping past the armature conductors, create in them very strong currents, and the mechanical reaction between these currents and the lines of the field tends to rotate the armature with great force. If the armature were movable it would thereby be set in rotation. But in the particular machine I am describing, and which, by the kindness of the Allgemeine Elektrizitaets Gessellschaft, of Berlin, I am able to show you here at work, the armature is fixed, whilst the field magnet, that is, the part shown in *Fig.* 37, can rotate. We have then a twisting couple between the armature and field; the armature cannot move, and, therefore, the field must move. Let us now see what is the effect of this movement. Say that the direction of the currents is such as to produce, when the central part of the machine is at rest, a clockwise rotation of the lines of force. The speed of rotation between the lines and the wires corresponds, of course, always to the frequency. If the wires are stationary, the lines revolve in relation to any fixed object in space (for instance, the wires of the armature) with the full speed given by the frequency, say, for instance, thirty revolutions per second if the frequency is thirty and our machine is wound for two poles, as shown in *Fig.* 37. Each wire of the armature will, therefore, be cut thirty times by a north field, and thirty times by a south field in each second, and the torque produced will set the central drum rotating counter-clockwise. Say, for instance, that the central drum runs backwards with a speed of twenty revolutions per second. The relative speed between the central drum and the lines of force is, of course, still thirty revolutions per second, but of these thirty revolutions, twenty revolutions are made up by the backward rotation of the drum, leaving only ten revolutions of forward speed for the lines of force in relation to any fixed point in space. The wires of the armature are now cut only ten times per second by a north field, and ten times per second by a south field. If we allow the drum to run faster still, the speed of cutting lines will be still further reduced. If, for instance, the central drum is so lightly loaded that it can acquire a speed of twenty-nine revolutions, the absolute speed of the field in relation to the armature will be reduced to one revolution, and each armature wire will be cut by a north field only once a second, and by a south field also once a second. You see, therefore, that the less the load on the motor the faster it will run, and this is precisely the same condition as obtains in an ordinary continuous current motor. At starting, when the drum is at rest, we have the greatest torque, and as the speed increases the torque diminishes. This is a very important property of the three-phase motor, since in consequence of it the machine not only becomes a self-starting motor, but one which will start with a large load. How large the starting load may be depends on the more or less skilful design of the motor. There are, as already pointed out, certain reactions of the armature on the field which tend to decrease the starting torque, but the subject is too difficult and intricate to be treated in the limited time at my disposal. I have merely given you a bare outline of the action of this class of machine, so that you may understand in a general way the principle of working.

The difference in speed of the drum and the field is technically termed the magnetic slip of the motor, and you will easily see that to obtain a small magnetic slip, and, therefore, a close approach to a constant speed, we must employ an armature of small resistance. Here, again, there is a close analogy between the three-phase motor and an ordinary continuous current motor with shunt or separately excited magnets. In practice, the magnetic slip need never exceed 10 per cent., and is generally between 3 and 5 per cent. This means that the speed of the motor only varies 5 per cent. between full load and no load.

In the machine which I have described, and which I can show you in action, the inner revolving part is the field magnet, but you will easily understand that the design could also be reversed by making the outer ring the fixed field magnet, and the inner drum the revolving armature. This arrangement is, in fact, adopted for small motors, because in this way we avoid altogether the necessity of using rubbing contacts, but it has the disadvantage of increasing the loss from hysteresis. I have shown you that the speed with which the field sweeps through the iron of the armature is very small, namely, that corresponding to magnetic slip, whereas the speed with which the field sweeps through the iron of the field-magnet is that due to the frequency, or about twenty times as great. The hysteresis loss in the armature is, therefore, trifling as compared with that of the field, and it is obviously of advantage to have less iron in the field than in the armature, which is done by making the inner drum the field, and the outer cylinder the armature. In small machines, where efficiency is not of paramount importance, the opposite arrangement is adopted, because of its greater simplicity and reduced cost.

The three-phase motor has several advantages over its two rivals, the ordinary continuous current motor and the ordinary alternate current motor, whilst, in a certain measure, it combines the good qualities of both. It is better than the continuous current motor, because of its greater simplicity. There is no commutator, and there are no brushes. There can be no sparking, and the motor may, therefore, safely be used in coal mines and other places where a machine that is liable to sparking would be dangerous. As a matter of fact, Mr. Tesla has already constructed motors for coal-cutting machines. Its greater simplicity, and more robust construction, renders it also applicable on board ship and other places where it is exposed to rough usage. It would, for instance, be perfectly feasible to design a three-phaser which will stand being drenched with sea-water, and yet work on as if nothing had happened. Another advantage is that the distance over which power has to be transmitted can be much increased. With ordinary continuous current motors this distance is limited, because we cannot make such machines, especially if of small power, for high voltages. With a three-phaser there is no such narrow limit to the voltage, for it is always possible to work through transformers, raising the voltage at the generating station, and letting it down again at the motor station, and this can be done with very small loss. Thus, in the Lauffen transmission of power, the voltage of the generating machine was only 50 volts (measured from any of the three terminals to earth), whilst the voltage of any line wire measured in the same way was 160 times as great in some experiments, and 320 times as great in others.

Ordinary alternators offer, of course, the same facility of transmitting power at high voltage and utilizing it at low voltage, but they do not offer the same facility for distributing the power in small parcels, because each motor must be provided with some source of independent electrical energy for starting and field excitation. It is also claimed by Herr von Dobrowolsky that the total weight of copper in the line is better utilized if arranged in three wires for the three-phase current than in two wires for a single-phase current, but on this point I cannot give you my own opinion, as I have not yet investigated it. One of the objections against the three-phase current is that it does not admit of a variable speed of motor, which, for many purposes, especially for traction work, is an absolute necessity. This, no doubt, is a serious drawback, but we may reasonably expect that the men who have succeeded in transmitting something like 200 horse-power over a distance of 110 miles will, in time, also succeed in solving this problem.

APPENDIX I.

Voltmeter absorbs in time T the energy—

$$wT = \int_0^T \frac{e^2}{r}\, dt.$$

$$wT = \frac{1}{\omega}\frac{E^2}{r}\int_0^T \sin^2(\omega t)\, d(\omega t).$$

$$wT = \frac{1}{\omega}\frac{E^2}{r}\left[\frac{\omega t}{2}\right]_0^T.$$

$$wT = \frac{T}{2}\frac{E^2}{r}.$$

And since

$$w = \frac{e^2}{r},$$

$$e = \frac{E}{\sqrt{2}}.$$

APPENDIX II.

Mean current, as determined by electrolysis, is Culomb's divided by time—

$$\frac{T}{2}\,\omega=\pi.$$

Mean current—

$$c=\frac{1}{\frac{T}{2}}\int_{0}^{\frac{T}{2}} i dt.$$

$$c=\frac{2}{T}\int_{0}^{\frac{T}{2}} I\ \sin\ (\omega t)\ dt.$$

$$c=\frac{2I}{\omega T}\int_{0}^{\frac{T}{2}} \sin\ (\omega t)\ d\ (\omega t).$$

$$c=\frac{2I}{\omega T}\Big[\cos\ \omega t\Big]_{\frac{T}{2}}^{0}.$$

$$c=\frac{I}{\pi}\times 2.$$

$$c=\frac{I}{\frac{\pi}{2}}.$$

The effective current is—

$$i=\frac{I}{\sqrt{2}};$$

therefore

$$c=i\,\frac{\sqrt{2}}{\frac{\pi}{2}},$$

or very nearly

$$c=0{\cdot}9\ i.$$

APPENDIX III.

Work done by current during one cycle is wT, and per second it is—

$$w = \frac{1}{T}\int_0^T eidt.$$

$$w = \frac{IE}{\omega T}\int_0^{2\pi} \sin \alpha \sin (\alpha + \phi)\, d\alpha.$$

$$w = \frac{IE}{2\pi}\int_0^{2\pi} \cos \phi \sin^2 \alpha d\alpha + \sin \phi \sin \alpha \cos \alpha d\alpha.$$

$$w = \frac{IE}{2\pi}\left[\cos \phi \left(\frac{\alpha}{2} - \frac{1}{2} \sin \alpha \cos \alpha\right) + \sin \phi \left(\frac{1}{2} \sin^2 \alpha\right)\right]_0^{2\pi}.$$

$$w = \frac{IE}{2\pi}\left[\cos \phi \pi\right] \text{ or } w = \frac{IE}{2} \cos \phi \text{ or } w = ie \cos \phi.$$

APPENDIX IV.

The power is $w = \frac{1}{2}$ IE cos ϕ, or substituting for cos ϕ the value $\frac{r}{\sqrt{r^2 + \omega^2 L^2}}$, and for I the value $\frac{E}{\sqrt{r^2 + \omega^2 L^2}}$, we have also—

$$w = \frac{1}{2} E^2 \frac{r}{r^2 + \omega^2 L^2},$$

$$w = \frac{1}{2} E^2 \frac{1}{r + \frac{\omega^2 L^2}{r}}.$$

The variable is r, and to find for which value of r the power w becomes a maximum, we resolve the equation $\frac{dw}{dr} = 0$, and find $r = \omega L$, and the maximum power—

$$w = \frac{1}{4} \frac{E^2}{r},$$

or, if by e we represent the effective voltage, such as would be indicated on a Cardew voltmeter, we have also—

$$w = \frac{e^2}{2r}.$$

The analogy with the well-known rule for maximum power from a source of continuous current is remarkable. According to this rule, maximum power will be developed in the external circuit, if its resistance is equal to the resistance of the battery or machine which gives the current. If E is the electromotive force of the battery, and r its internal resistance, the maximum power which is obtainable in an external circuit of equal resistance is—

$$w = \frac{1}{4}\frac{E^2}{r},$$

precisely the same expression as obtained above for alternating currents.

APPENDIX V.

Let ϕ be the angle of lag in the circuit, the power given to which is to be measured, and let δ be the angle of lag in the fine wire coil of the wattmeter, due to its self-induction. Let I be the current through the thick wire coil, and i the current through the fine wire coil, then the power indicated by the wattmeter, if the currents were steady, would be $KriI$ where K is the coefficient of the instrument, and r the resistance of the fine wire coil. The true watts of the alternating current of E volts are—

$$W = IE \cos \phi.$$

The indicated watts are—

$$W' = K \text{ (Reading)}.$$

$$W' = Kri \cos (\phi - \delta)\, I = KE \cos \delta \cos (\phi - \delta)\, I.$$

Therefore, to get true watts, we must multiply the watts indicated by

$$\frac{\cos \phi}{\cos \delta \cos (\phi - \delta)},$$

which expression can also be written in the form—

$$\frac{1 + \tan^2 \delta}{1 + \tan \delta \tan \phi}.$$

If the wattmeter has no self-induction, $\delta = 0$, and no correction is required. Again, if the self induction of the wattmeter is equal to that of the circuit to be measured—

$$\tan \delta = \tan \phi, \text{ and } \frac{1 + \tan^2 \delta}{1 + \tan \delta \tan \phi} = 1.$$

In this case also no correction is required. In all other cases the corrections given in this formula must be applied.

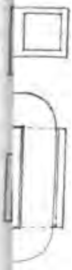
TRANSFORM

PLATE, I.

utput of all 6 K.W.

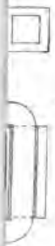
Scale, 1/2" = 1 Foot.

Iron.	100 lbs.
Copper.	340 „
Hysteresis.	1·45%



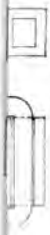
Iron.	105 lbs.
Copper.	93 „
Hysteresis.	1·53 %

ron.	105 lbs.
opper.	120 „
ysteresis.	1·57%



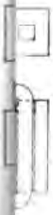
Iron.	114 lbs.
Copper.	48 „
Hysteresis.	1·7 %

n.	119 lbs.
per.	54 „
teresis.	1·74%



Iron.	135 lbs.
Copper.	27 „
Hysteresis.	1·94 %

n.	154 lbs.
per.	25 „
steresis.	2·2 %



Iron.	173 lbs.
Copper.	15 „
Hysteresis.	2·55 %

Iron.
Copper.
Hysteresis

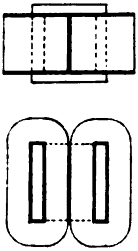
ron.
opper.
ysteresis. 1

on.
pper.
ysteresis. 1

n. 76
per. 23
steresis. 1

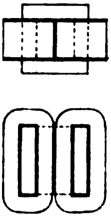
PLATE. II.

Iron.	75·5 lbs.
Copper.	290 „
Hysteresis.	1·11%



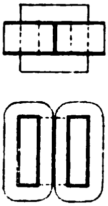
Iron.	83 lbs.
Copper.	217 „
Hysteresis.	1·53%

ron.	89 lbs.
opper.	94 „
ysteresis.	1·28 %



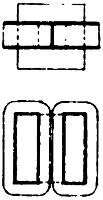
Iron.	98 lbs.
Copper.	72 „
Hysteresis.	1·44 %

on.	102 lbs.
pper.	57 „
steresis.	1·42 %



Iron.	111 lbs.
Copper.	44 „
Hysteresis.	1·58 %

n.	78 lbs
pper.	23 „
steresis.	1·8 %



Iron.	138 lbs.
Copper.	29 „
Hysteresis.	2·03 %

PAPER IV.

THE APPLICATION OF WORKS TO IRREGULAR GROUND.

By Capt. S. D. Cleeve, R.E.

(Compiled for use of Cadets of the Engineer Division, R.M. Academy, Woolwich, and published by request of several brother officers).

Object.—The general idea of, or objects to be fulfilled by, a work or parapet, field or permanent, being given, the position on the ground combining the most advantages has to be accurately selected for the same, and the subject, though essentially practical, involves certain theoretical principles of geometrical construction. The object aimed at is twofold: (1), to adapt the crest line or parapet to the site in such a manner that the greatest advantage is secured for the defender's fire over the whole of the enemy's possible approach; (2), at the same time to thoroughly defilade or screen the interior of the work from the enemy's *view* in any position he might be able to take up, and as far as possible from his *fire*.

(A). *Field Works.*—For all field or hasty works of fortification, the above objects are arrived at practically in the field on the spot by "trial," and testing different positions by taking lines of sight with the aid of pickets, pieces of string or tapes, men carrying poles, etc.; this is fully treated of in the standard text-books of field fortification (see *Defilade*, etc.), and is not, therefore, dwelt upon in these notes. If the explanations and principles given in the text-books under the above head are thoroughly understood and appreciated, it will be seen that, in dealing with works of this class, a contoured map of the site and surrounding ground is not required.

(B). *Permanent Works.*—In permanent works the case is very different, and may be better described as the theory of adapting designs of works to suit ground given on accurately-contoured plans—the adaptation of the several planes of defilade or fire to the best advantage. Here the Engineer officer finds himself at what may be said to be one of his special provinces, whereas what is commonly spoken of as practical defilade in the field (A) should be mastered by all who study even the elements of field fortification.

To enable anyone to attempt (B), it is absolutely necessary to be able to read easily any contoured map or plan, and to thoroughly understand what are commonly called the simple problems on solid geometry on the index system of projection. This knowledge will be assumed in the following notes.

It being generally acknowledged that by practice and experience alone can the subject be mastered, and that it is impossible to formulate all the various conflicting conditions which may present themselves, it is proposed here—

(I.). *Definitions and Principles.*—To endeavour to explain some of the difficulties which meet every student at first, and to lay down a few principles which experience has shown are seldom grasped by those who find themselves face to face with their first example to work out.

(II.). *Examples.*—To work out a few examples of some of the most ordinary cases that occur relating to ground outside the work.

(III.). *Arrangements for Interiors.*—To offer a few hints on the arrangement of interiors.

When the notes and examples under these three heads have been thoroughly grasped, the student ought to be able to deal with any ordinary project of defilade or design.

(I.).—DEFINITIONS AND PRINCIPLES.

(*a*). *Commanding Point.*—Any point outside a work on a higher level than the crest would be called a "commanding point," with respect to that crest. One point would be said to be "*more* commanding than another," with respect to an assumed point on a crest line, when it has a greater angle of elevation from the point on the crest line than the other point has.

N.B.—A *commanding point* has sometimes been called a *point of danger.*

(*b*). *Tangent Plane* (see *Fortification* text-book) is an imaginary plane tangent to the ground at the enemy's position, and passing a certain number of feet (usually the assumed command of the crest line *less* the command of enemy's work) above the ground at the gorge of the works to be defiladed.

(*c*). *Plane of Defilade* (see *Fortification* text-book) is an imaginary plane passing through the crest line of the enemy's work, and the crest line of the work to be defiladed. It is therefore parallel to the "tangent plane," and on the above assumption would be (vertically) the height of the command of the enemy's work above it.

N.B.—Considering the probable distance between any work and an enemy's position, the difference between (*b*) and (*c*) may often be neglected.

(*d*). Ground is often said to be defiladed for a height of x feet at any point when the plane or line of fire from the crest line at its greatest depression does not pass more than x feet above the ground at the given point.

(*e*). Viewed in any direction from any point on a crest line, the ground in front is obviously unseen in the following cases:—

(1). When it falls below the plane of the superior slope.

(2). If a tangent line is drawn at a less inclination than the superior slope to any feature of the ground, then the ground beyond the point of contact is invisible until it again rises up to, or above, the tangent line.

Such ground would be treated as undefended by fire from this crest, but might, of course, be defended by the fire of an adjacent work or by curved fire.

From any point in a crest line there may be several tangents to an irregular surface all in the *same direction*, but at *different inclinations*, less than that of the superior slope. In any such case all these tangents would have their plans in the same straight line. Two illustrations (*Plate* I., *Figs.* 1 and 2) are given, and must be thoroughly mastered.

(1). "Looking up" at a surface of irregular ground (*Plate* I., *Fig.* 1).

(2). "Looking down" at a surface of irregular ground (*Plate* I., *Fig.* 2).

In each case several (here *three*) lines are drawn in the same direction, but at different inclinations, and each is a tangent to the ground at one point. In case (1) out of these three lines the line $p'b'$ is *the* tangent to the ground from the point p', within the limits of the drawing; it will be observed that this line $p'b'$ is that tangent, of all those drawn, which has the *greatest* angle of inclination, because any other of the tangents must cut the ground surface somewhere within the limits of the figure. By similar reasoning, in case (2) the tangent $p'f'$ of *least* inclination is *the* tangent to the ground "looking down." Unless the points a', b', c' (*Fig.* 1), and d', e', f' (*Fig.* 2), represent in section level straight ridges, which could not well occur, there will evidently be many series of tangent planes containing the lines $p'a'$, $p'b'$, $p'c'$, etc.

(*f*). If, instead of regarding the crest as a point, we deal with it as a fixed line, any number of tangent planes can be drawn containing that line, but only one point in the ground, or, in other words, an imaginary plane rotating about the given line, may be a tangent plane to the surface, in different positions at several different points. It may be necessary to select *the* tangent plane to the surface (containing the given crest point or line) which best fulfils the requirements of the project under consideration.

A plane, containing a given fixed crest line, may, subject to the limit introduced by the superior slope, have any inclination between 90° and the inclination of the line itself. When in its position of maximum inclination (90°), all the horizontals of the plane coincide in plan with the plan of the line. When the plane is in its position of minimum inclination, *i.e.*, the same inclination as the line itself, its horizontals are perpendicular to the plan of the line.

Let a_0b_{40}, be the plan of a line (*Plate* I., *Fig.* 3). It is required to obtain tangent planes containing a_0b_{40} to a surface not shown in the figure, but lying on the "left" of the line a_0b_{40}. Graduate a_0b_{40} at intervals of 10 units for convenience, the ground (not shown) being contoured at the same levels.

From any point in ab, conveniently c_{20}, draw cx, representing the plan of the horizontal (20) of a plane containing ab, and rotating about it, and being *a tangent plane* to the surface in the direction indicated by the arrow; this line, cx_{20}, must therefore be tangent to the ground at some point on the contour of 20 on the ground. If we imagine different successive positions of cx in order from ca round to cb, *i.e.*, in order of hands of a clock, the tangent plane of which cx is the horizontal, which we will call, for brevity, "the tangent plane abx," is vertical, *i.e.*, a *maximum*, when cx coincides in plan with ca *i.e.* has the same inclination as the line a_0b_{40}, or is a *minimum* when cx is perpendicular to ca_0, and again is vertical, *i.e.*, a *maximum*, when cx coincides in plan with cb.

In the passage of *cx* through the first quadrant in the diagram, *i.e.*, as "the tangent plane *abx*" decreases from its maximum to its minimum, we are said to be "looking up" at the surface (*Plate* I., *Fig.* 1), and in passing through the second quadrant, *i.e.*, as "the tangent plane *abx*" increases again, we are said to be "looking down" at the surface (*Plate* I., *Fig.* 2). Now for tangent planes "looking up" we must select from all our trials the "steepest," and from those "looking down" the most gentle one.

If the plans of two or three different positions of "*cx*" in each quadrant are assumed, it will be seen without difficulty that we shall ascertain the plane required by selecting in either case that position of the line *cx*, representing "the tangent plane *abx*," which in plan makes the smallest angle with ca_0. For the first quadrant this will be an *acute* angle, and will give us the "steepest" plane, and for the second quadrant an *obtuse* angle, giving us the plane of "least inclination" of all those from which our selection has to be made.

Or we may state the general rule thus:—

"To find *the* tangent plane passing through a given line, and tangent to a given irregular surface." Graduate the line at similar intervals to the contours of the ground; draw lines from each index on the line tangential to contours of corresponding indices of the ground; each of these tangents will be a horizontal of *a* plane tangential to the ground, and containing the given line; we now select whichever tangent makes the least angle in plan, whether acute or obtuse, with the lower part of the given line. If other horizontals are drawn through the other indices on the given line parallel to this tangent, we obtain the scale of slope of the plane containing the given line and the selected tangent. This will be the scale of slope of the required tangent plane to the ground.

(II.).—EXAMPLES.

(1).—A Crest Line being given to Determine a Plane Containing the Given Crest Line, which shall be "the" Tangent Plane to an Irregular Surface given by its Contours.—(Plate II.).

Let *hk* represent the trace of the crest line of the left flank of a proposed work; it is required to find scales of planes containing this crest line, and tangential to the ground shown by the contours.

Produce *hk* to *o*, and graduate the line *ho* at similar intervals (l_{50}, m_{45}, n_{40}) to the contours of the ground.

Let us first deal with that part of the ground which lies more directly in front of *hk*, and is furthest from it on the plate. Following the rule given above (definitions and principles), draw lines from *k*, *l*, *m*, *n*, each tangential to the contour of its own level on the ground, *i.e.* :—

k_{55} a_{55}
l_{50} b_{50}
m_{45} c_{45}
n_{40} d_{40}

Each of these lines is one horizontal of *a* plane containing the crest line *hk*, and touching the ground. Select that one which makes the least angle in plan with the lower part of the line *ho*. Thus the horizontal $l_{50}b_{50}$ is chosen, the angle *olb* being less than either *ond*, *omc*, or *oka*, as the horizontal of level (50) in the required tangent plane to the ground. The scale of the plane is then obtained by drawing lines through *hkl*, etc., parallel to *lb*, and the result is figured and marked, as in Scale A, *Plate* II.

Having obtained the above result for the part of the ground surface first considered, it is evident that other tangents to contours might have been drawn from *l*, *m*, *n*, to another part of the ground surface (nearer to the given crest line), as :—

l_{50} p_{50}
m_{45} q_{45}
n_{40} r_{40}

From this new set of tangents, the one, $m_{45}q_{45}$, would give *the* tangent plane (figured and marked Scale B) to this part of the ground surface. Thus altogether *two* tangent planes to the given ground have been obtained, each containing the crest line of the flank of the proposed work—and each being, for separate parts of the ground surface, *the* tangent plane to that part of the surface within the limits of the drawing.

It is evident that for purposes of design or trial in dealing with a real project of fortification, either or both of these tangent planes might be required for some special purpose.

(2).—*To Determine the Form and Extent of Ground unseen from a Given Point.—(Plate III.).*

(In other words, to find successive tangent lines to the ground from the given point. There are three methods of doing this:—

(*a*), by an application of the previous example; (*b*), by trial and elimination; (*c*), by sections).

In this example any limit to depression introduced by the superior slope is left out of consideration.

Let p_{70} be the plan of a given point. It is required to find what portion of the ground, represented by the given contours, is unseen from the point *p*, within the arc *apd*.

In *Plate* III., for simplicity, two plans of the same piece of ground are given (*Figs.* 1 and 2), and three different methods of solving the problem are shown. *Fig.* 1 illustrates the tangent plane method and that of trial and elimination. In *Fig.* 2, the further limit of the unseen ground is first obtained (the nearer boundary having been transferred from *Fig.* 1), and sections are used, illustrating the third method of solving the whole problem.

It is evident that if within the arc *apd* the plans of any number of lines be drawn from *p*, such as *pb*, *pc*, we can find where each of these lines touches the ground, and also where each line, if produced beyond such point of contact, intersects the ground surface. By joining all these points of contact and points of intersection on each successive line in order, the whole of the ground included between these curves will be unseen from *p*. The greater the number of radial lines drawn from *p*, the more accurate will be the boundary of the unseen ground. In the given case only four lines in all are assumed, viz., *pa*, *pb*, *pc*, *pd*.

In *Fig.* 1 the following methods are employed:—

(*a*). The tangent plane method for the two lines *pa*, *pd*.

(*b*). Trial and elimination for *pb*, *pc*.

(*a*). Where *pa*, *pd*, in plan cross each contour, letter them $a'a''a'''$, $d'd''d'''d''''$. From the point p_{70} draw a line *px*, conveniently, as in the figure, and graduate it at similar intervals to the contours of the ground, *i.e.*, h_{60}, k_{50}, l_{40}, m_{30}, n_{20}. Dealing first with the line *pa*, let pa' be the plan of a line from *p* meeting the ground at a', pa'' meeting the ground at a'', pa''' meeting the ground at a'''. It is evident that whichever of these lines has the *least* inclination will represent to us the tangent line from the ground surface (as shown by the contours) from the point *p*, in the direction *pa* (see *Fig.* 1, *Plate* I.).

Join $(a'k)_{50}$, $(a''l)_{40}$, $(a'''m)_{30}$. Imagining auxiliary planes containing the line *px*, and each of the points a', a'', a''', separately, these lines would respectively represent one contour of each auxiliary plane. The relative inclinations of these auxiliary planes afford a certain measure of the relative inclinations of the lines pa', pa'', pa'''. By the previous example we can at once select $(a''l)_{40}$ as the horizontal of the auxiliary plane of least inclination, and therefore pa'' will be the plan of the line of *least* inclination from p. In other words, *the* tangent line to the surface from p, in the direction pa, will be the line $p_{70}\ (a'')_{40}$.

If pd be treated similarly to pa, then by analogous reasoning $p_{70}\ (d''')_{30}$ is the line of least inclination from p, in the direction pd, and tangent to the ground at the point d'''. We could deal with pb, pc, or any other radial lines from p in like manner, and find any number of points of contact of any number of tangents from p to the surface of the ground.

(b). Considering pb, we *assume* that b' is the point of contact of the tangent to the surface; find the scale of the line $p_{70}\ (b')_{40}$ (by trial with a pair of dividers*). From b' downwards set off this scale, and if, in so doing, each index on the line pb falls below the similarly figured contour of the ground, it shows us that the line we are scaling does not intersect the ground surface anywhere, but is keeping above it, and therefore b' must be the point of contact of *the* tangent line to the ground through the point p_{70}. Adopting the same method on the line pc, assume c' to be the point of contact of the tangent from p. Finding the scale of $p_{70}\ (c')_{50}$, and proceeding as above, we find that the next interval (level 40) on the line pc, below c', falls *above* the contour of its own level 40 on the ground, and, therefore the line $p_{70}\ (c')_{50}$ must penetrate or cut the surface, and cannot, therefore, be the tangent to it.

Repeat the above process, assuming c'' to be the point of contact of the tangent from p. Graduate $p_{70}\ (c'')_{40}$, and by scaling this line with the dividers from c'' downwards, we find that the line keeps above the surface of the ground everywhere, and therefore c'' is the point of contact of *the* tangent from p_{70}.

In *Fig.* 2, *Plate* III., a larger extent of the same piece of ground is given by the contours, and the nearest boundary of the unseen portion, $a''b'c''d'''$, is transferred from *Fig.* 1. We now proceed to find the further limit of this unseen area, which simply means to ascertain where each of the tangent lines pa'', pb', pc'', pd''', *intersects* the ground.

(a). This may be done by imagining the ground to be a uniformly plane surface between each pair of contours, where it appears to be likely that the tangent line from *p* produced should intersect it. Referring to *pb*, assume any auxiliary plane (20 and 30 horizontals shown by dotted lines) containing the line $p_{70}\ (b')_{50}$. Intersect this plane with that of the ground, giving us the point *q* as the intersection of the tangent line *pb'* with the ground surface. Similarly for each other radial line from *p*; *o*, *q*, *r*, *s* are found. Join them in order, thus obtaining the further limit of the unseen area.

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* **A piece of elastic is often useful for this purpose.**

(*b*). Sections may be used to solve the problem. Four sections are shown on the plate (any assumed vertical scale might be used), and the points of contact and those of intersection of any tangent lines to the surface in each section are readily obtained, and transferred to the plan of the ground.

The above example deals with a piece of ground, the unseen portion of which would be in one area on the plan. In practice, however, we are much more likely to find that the unseen ground from any given point would consist on the plan of several disconnected patches. For instance, in any one direction, such as *pa* (*Figs.* 1 and 2, *Plate* III.), we might find such undulations in the ground as would give us more than one unseen portion of the same, as viewed from *p*.

This is best illustrated by *Fig.* 2, *Plate* I. Here *pf* is *the* tangent to the surface; but *pd* and *pe* must also be taken into consideration if we are dealing with the question of *unseen* ground, because each of them will define for us a separate portion of ground unseen from *p*.

The case may generally be stated as follows:—

In finding unseen ground from a point in any direction, *first* find *the* tangent to the surface in that direction. Then *all* ground *beyond* the point of contact of this tangent *must* be unseen from the given point. There may, however, be other portions unseen, and lying between the point of contact of *the* tangent to the surface and the given point, and this can be ascertained by finding *the* tangent to this portion of the ground surface only (should there be one), as *pe* in *Fig.* 2, *Plate* I. Then, looking back towards *p*, considering the portion *pe*, we find it has a tangent at *d*, giving another piece of unseen ground just below *d*.

(3).—*To Find the Tangent Plane to Two Prominences Given by their Contours.*—(*Plate IV.*).

(An application of Example 1, solved by trial and elimination).

Let p_{20} be the plan of the given point. It is required to find the scale of a plane passing through the point p, and tangent to the two prominences represented by the contours.

It is evident that a line through p, in the direction of pq for convenience, anywhere between the prominences, *if correctly indexed*, would be the plan of some line in the required plane, which would ascend more or less from p in the direction of q. Moreover, the horizontals of the required plane would pass in plan through points in pq of their respective levels, and *at least* one of these horizontals must be a tangent to the contour of its own level on each prominence.

> N.B.—One horizontal may touch the contour of its own level on both prominences, but the condition is generally fulfilled by at least two different horizontals of the tangent plane being tangents one to each prominence.

Graduate the assumed line pq at similar intervals to the contours of the ground, and as near as can be judged, by inspection at the slope that the required tangent plane might have, *i.e.*, from 20 to 60 at 10 units interval.

> N.B.—A little practice, and an accurate conception of the ground represented by the contours, should enable anyone to make a very near approximation to the correct result in one or two trials.

With this assumed line p_{20}, q_{60}, by an application of Example 1, find the scales of two planes passing through pq, and being, one on each side of pq, *the* tangent plane to the prominence on that side, *i.e.*, to one at a and the other at b. Call the scales of these two tangent planes (A) and (B). We now have a tangent plane to one prominence, Scale A, passing through pq, and a tangent plane to the other, Scale B, also passing through pq.

It should now be easily seen that *if* these two tangent planes gave parallel and equal scales, the problem would be solved, and we should have found *one plane*, since Scale A and Scale B would then represent one and the same plane, which would pass through p, and be tangent to the two prominences.

Should Scale A and Scale B, in the first attempt, not prove equal and parallel to each other, as is the case in this example, it follows that the assumed line pq was *not* in the required tangent plane to the two prominences. The process must, therefore, be repeated, graduating pq at a different slope until that particular slope of pq is obtained which will bring the *two* scales (A and B) of the tangent planes equal and parallel.

A second trial is shown on the plate, the line pq being re-graduated from *p* at the points shown, giving us trial planes (Scale A′ and Scale B′), shown by chain-dotted lines, and tangent to the prominences at *c* and *d* respectively.

It will be seen that although this attempt is a nearer approximation to the required result than the first one, the Scales A′ and B′ are not equal and parallel.

A third attempt must, therefore, be made, as shown on the plate, and the re-graduation of *pq* gives us the *one* scale C, which is tangent to one prominence at *h* and the other at *k*.

(4).—*To Find the most "Commanding Points" of Surrounding Country with Respect to an Assumed Position (a Point Centrally Placed on the Plan).—(Plate V.)*

(Solved by graduating a line as generatrices of a series of inverted right cones, with the assumed point as vertex, and by ascertaining the intersection of these cones with the ground).

Let p_{50} be the plan of the given point. It is required to find, with respect to *p*, the most commanding points on the ground shown by the contours, or to compare the relative commands of any points we may select.

It is evident that the most commanding points on any contour will probably be somewhere on a *convex* portion of the contour as viewed from *p*. Assume any point *a* as one such point on the highest (or any other) contour; join p_{50}, a_{250}; graduate this line at similar intervals to the ground contours, and consider it as the plan of a generatrix of a vertical inverted right cone, with *p* as vertex. The contours (at similar intervals to the ground) of this cone would be the chain-dotted arcs on the plate. If we should find that the surface of the ground does not touch or intersect this conical surface *anywhere*, *a* would be the *most* commanding point on the plan. On examination of the plate, however, we find that in the vicinity of *c* and *d* the ground does intersect the conical surface, and, therefore, in each of these places is "more commanding" with respect to the point *p* than the ground at *a* is. We also observe that the ground at *b* *touches* the conical surface, but does not intersect it; the ground at *b* has, therefore, the same command as that at *a*.

We have next to compare the relative commands of ground near *c* and *d*. Proceeding as in the first instance, join *pc*, and contour the cone traced by *pc* as generatrix. These contours are shown by broken lines. We now observe that the ground at *d* does not touch or intersect this cone, and, therefore, has less command than the ground at *c*.

So far, therefore, we have found four points on the ground, each commanding the given point *p*, in the following relative order:—

Point of greatest command	...	*c*.
„ next „	...	*d*.
„ least „	...	*a* & *b*.

(5).—*To Artificially Obviate Undefended Spaces in Front of a Crest Line, or to Apportion the "Remblai" and "Deblai" of a Glacis.*—(*Plate VI.*).

[This is chiefly a question of trial. Glacis planes are assumed passing through the crest line, and their curves of intersection with the ground traced. Regarding any such plane as the plane of sight from the crest line, then ground rising above this plane may obscure view beyond, and would have to be cut away. All ground below the plane would be unseen, and may form hollows in which an enemy might get cover, and these should, if possible, be filled up (though, of course, such ground might be defended by other means). If the hollows are not deeper than three feet, they are often considered to be defended].

N.B.—That plane should, theoretically, be selected which, when adopted as the plane of sight, leaves no ground in the neighbourhood of the work unseen. Practically, however, it is best to select a plane which, while leaving a minimum of ground unseen, involves no excessive amount of *remblai* or *deblai*, and gives approximately equal quantities of both. The distance from the work within which unseen ground is permissible is a practical question to be determined for each case on its own merits; the *remblai* or *deblai* should rarely exceed 10 feet in depth.

Let $a_{75}b_{80}$ be the plan of a crest line on the brow of the slope represented by the contours:—

It is required to find the most suitable plane for a glacis in front of the given crest line.

Dealing only with that portion of the ground included between the two chain-dotted lines from *a* and *b*, and assuming that the general slope of this ground would not demand a greater depression of fire from the crest line for its general defence than is practicable the problem may be re-stated thus:—

It is required to find the scale of a plane passing through $a_{75}b_{80}$, and intersecting the given ground in such a manner that all earth above the plane, *i.e.*, *deblai*, should, in volume, approximately equal (or if we allow for increase on excavation, should be proportionately less than) all hollows or depressions below the plane which would have to be filled up (*remblai*) in order to raise such portions of the ground into the plane of fire. The *deblai* portions being removed, as obscuring vision and fire from the given crest line, would provide earth for the *remblai*.

The problem is solved by trial. Assume a plane estimated to give approximately equal amounts of *remblai* and *deblai*. This may be done either by passing a plane of some assumed slope through *ab*, or by assuming some distant point on the ground, and finding the plane containing this point and the crest line.

In the example (*Plate* VI.) a point c_{40} is assumed, and the Scale B of the plane containing c_{40}, and the line $a_{75}b_{80}$, is found. This plane intersects the ground in the curved lines shown dotted in the plate, and it is not difficult to see that the ground we should have to remove, as being *above* the plane (B), would appear to give *more* than enough earth to fill up the hollows *below* the plane (B). We therefore make a fresh trial, and assume another plane of rather less inclination, *i.e.*, that through *d*, giving us the chain-dotted intersections with the ground surface, and what are more likely to prove approximately equal amounts of *remblai* and *deblai*.

Of course, in an actual design, the volumes of the *remblai* and *deblai* would be estimated, and attention must be paid to the note given above.

In the present case the glacis for the length of crest line under consideration is designed in one plane; but in the next example (5*a*) another method is given, which is sometimes more convenient, and in which the glacis is broken into two planes.

(5*a*).—*To Arrange to the Best Advantage the Planes of the Glacis of a Permanent Work when the Slope of the Natural Ground in Front of the Glacis Crest Line is Fairly Uniform and Approximates to that of the required Glacis.—(Plate I., Fig. 4).*

(In this case the balancing of *remblai* and *deblai* need not be considered except in the field).

From each angular point, or most important salient; in the plan of the crest of the glacis, determine and graduate a line following approximately the general fall of the ground straight to the front.

Assuming that each of these lines meets the ground (giving a point which will be in the intersection of the plane of the glacis with the ground), we have to arrange to the best advantage the glacis slope in front of each portion of the crest line contained between every pair of "graduated lines."

Let $a_{40}b_{30}$ be a portion of the glacis crest, and $a_{40}c_0$, $b_{30}d_{10}$ the graduated lines following roughly the slope of the ground (not shown), but ultimately meeting it at c and d; join the lower end, c_0, of the steepest of these lines with the upper end, b_{30}, of the other line. Then the plane containing *abc* must be steeper than that through *bcd*, and the whole of the glacis in question, *i.e.*, *acdb*, will be visible from the crest line *ab*, and made up of the two planes *abc*, *bcd*, intersecting each other in the line *bc*.

This method is a convenient one in designing a glacis on irregular ground; but it must be clearly appreciated that, in actually constructing works from approved designs which must necessarily be contoured and carefully calculated, all surfaces would be left in undulations rather than well-defined "ridges and furrows," as in the contoured designs of the same.

The geometrical accuracy of the draughtsman's design must be the first thing; the actual construction of that design should at once suggest the surrender of much of the finished details of the surface planes.

(III.).—THE ARRANGEMENT OF INTERIORS.

In the above examples the interior arrangements of a work have been left out of consideration. It is now proposed to give a few hints under this head (see object 2, page 131), *i.e.*, the screening of the interior of a work from the enemy's *view* (and as far as possible from his *fire*).

(1). *Protection from Fire.*—Protection from *fire* is of two kinds: (*a*), parapets and traverses; (*b*), overhead protection, casemates, bombproofs, and blindages.

With the exception of field guns in certain cases, overhead cover must be relied upon to provide protection against artillery fire, especially that of howitzers and mortars. As a rule, only a very limited amount of such overhead cover can be provided in the interior of a work, and how far this question should be considered does not come within the province of these notes.

(2). In a well-designed work, however, a great deal may be done to provide some protection against shrapnel, machine-gun, and long-range musketry fire. The lowering of the terrepleins or interior communications, the introduction of parados (occasionally), and in some special cases, as on the flanks of a work, the use of traverses, all suggest themselves as means to this end; but here again, as above, each particular case must be carefully considered on its own merits, and no hard-and-fast rules can be laid down.

(3). *Protection from View.*—Without in any way losing sight of the above points, the interior of a work should be so designed as to give complete protection to the garrison from the enemy's *view* in any position he might be able to take up. Here we come more within the province of these notes, and as no two cases would be likely to involve exactly the same principles in their design, rather than work out any assumed case the following hints may be useful in defilading from *view*, and in some cases from shrapnel, machine-gun, or musketry *fire*:—

(*a*). Ascertain the elevation and distance of the most commanding position or positions any enemy might take up (Example 4 may be useful for this purpose).

(*b*). Consider the angle of descent of artillery, machine-gun, or musketry fire which any assailant might be able, in any possible position, to bring to bear on the work (see table, page 147).

N.B.—We must remember that, viewed from one point inside a work, a certain angle of descent of fire over a crest line from one direction may be *less* dangerous than a much less angle of descent might be from another direction—depending, of course, upon the levels of the crest line, and the relative distances of the assumed point within the crest line in each direction.

(*c*). Decide how much protection to men moving about in the interior of the work it is desirable to give from the enemy's *view*, and, if possible, from his musketry fire.

(For example, we might assume that the communications inside the work should be kept at least seven feet below the enemy's plane of vision, or below his fire grazing the crest in some assumed direction at a certain angle of descent).

(*d*). We shall either have to consider the enemy's position as a point (or points), or as a line (or lines). If the former, we may deal with his plane (or planes) of vision as passing through any assumed *line* in or near our work; if the latter, we can only consider the plane (or planes) of vision through a *point* in or near the work.

All projects of designs we are likely to meet with may be classified under two heads:—

(I.). *When the work can be commanded or overlooked by the enemy.*

(II.). *When it cannot be commanded or overlooked by the enemy.*

Taking these cases in turn, the following general remarks may be useful:—

(I.). In a proposed design, having drawn the plan of the crest line, and having carefully considered each of the above points, we can then proceed to deal with the interior by assuming either a point or a line at a convenient maximum command, within the work, near the gorge, on the gorge, or in rear of the work clear of the gorge. Find the scale of the plane containing this point or line, and the enemy's position (line or point). This would be the plane of the enemy's vision, and by ascertaining the levels of the crest line of the work, *in this plane*, the interior will be hidden or defiladed from view of the enemy.

> N.B.—Although the distinction between "tangent plane" and "plane of defilade," as defined in the standard text-books, is here ignored in dealing with practical defilade, the principle remains the same, and it is not probable that the distinction between the two planes will affect future designs for permanent works.

Having thus ascertained the levels of any number of points in the crest line, we must next of all consider whether such levels would give an excessive command to the work; if so, the difficulty might be overcome by reducing the command of the originally assumed point or line, and by *excavating* the interior of the work, where necessary, so as still to preserve the required amount of cover from view, below the plane of the enemy's vision; or by introducing a central parados, in which latter case we would treat the portion of the work on the reverse side of the parados by itself, and, *assuming a new point or line*, on the exposed side of this parados another plane of enemy's vision would be found, and new levels obtained (in this plane) for that portion of the crest line of the work which lies between the parados and the enemy's position.

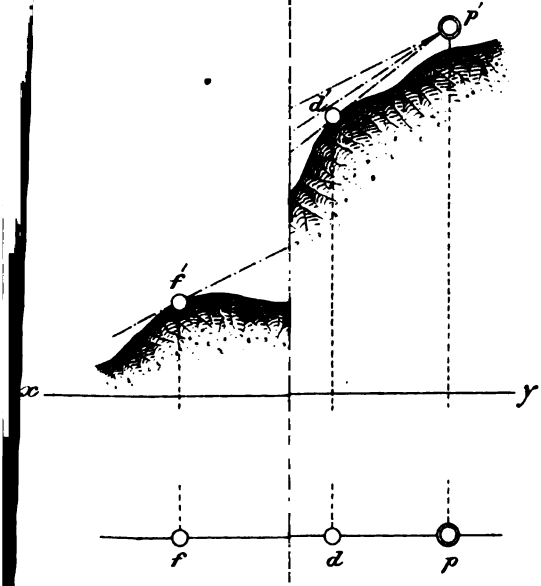
(II.). In the above case it is assumed that in some measure the enemy's position commands the proposed work, and that he can "overlook" the proposed site; this *may* occur, and we are bound to give such a case our first consideration, but in the original selection of a site the advantage of command would, if possible, be secured for the work, in which case the problem will be simplified. Having carefully considered (*b*), trial levels would be *assumed* for the crest line, and *lines* passed at certain slopes (representing enemy's artillery, musketry, or machine-gun fire) through *points* on the crest, or *planes* through lines of crest. By excavation, if necessary, the interior communications would be sunk to a safe depth below such lines or planes of fire.

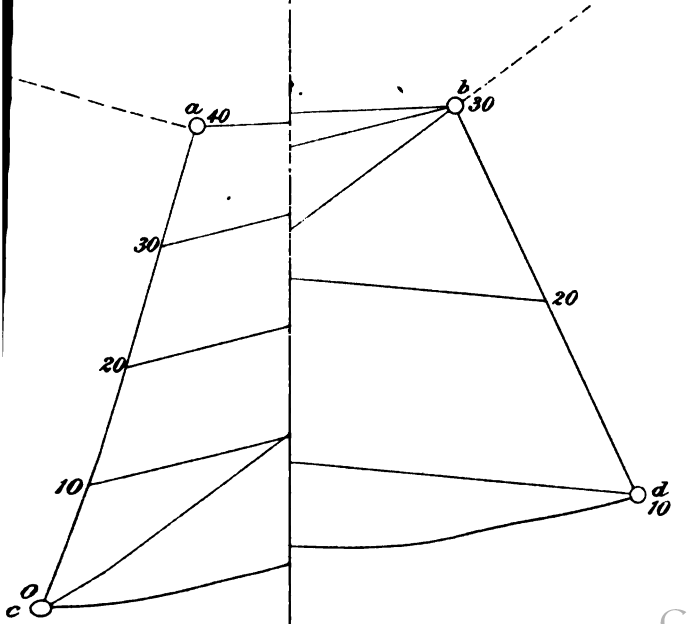
Table showing Approximate Angles and Slopes of Descent.

Range (yards).	Nature of Gun, etc.															Remarks.
	5" B.L.			2·5" R.M.L. (Steel Gun).			12-pr. B.L.			3-pr. Q.F. Nordenfelt.			Martini-Henry Rifle.			
	Angle of Descent.		Slope of Descent.	Angle of Descent.		Slope of Descent.	Angle of Descent.		Slope of Descent.	Angle of Descent.		Slope of Descent.	Angle of Descent.		Slope of Descent.	
	°	′		°	′		°	′		°	′		°	′		
500		29	1 in 118		56	1 in 62		35	1 in 98		30	1 in 114	1	22	1 in 41·9	
1,000	1	7	1 ,, 51	2	8	1 ,, 27	1	28	1 ,, 39	1	20	1 ,, 42	3	52	1 ,, 14·7	
1,500	1	56	1 ,, 29	3	49	1 ,, 15	2	36	1 ,, 22	2	27	1 ,, 23	8	20	1 ,, 6·8	N.B.—For shrapnel, the angle of descent at any range given in this table must be increased by half the angle of the cone of dispersion, which is as follows:—2·5" R.M.L. gun—Cone of dispersion, 14°; 12-pr. B.L. gun—Cone of dispersion, 16°. Corresponding slopes for any range can thus be calculated as required.
2,000	2	58	1 ,, 19	5	43	1 ,, 10	4	5	1 ,, 14	4	1	1 ,, 14	15	40	1 ,, 3·5	
2,500	4	10	1 , 13	8	8	1 ,, 7	5	43	1 ,, 10	5	59	1 ,, 9	27	50	1 ,, 1·8	
3,000	5	28	1 ,, 10·4	11	19	1 ,, 5	8	8	1 ,, 7	8	28	1 ,, 6	46	45	1 ,, ·9	
3,500	6	54	1 ,, 8·3	14	3	1 ,, 4	10	19	1 ,, 5·5	11	34	1 ,, 4·8	...		...	
4,000	8	35	1 ,, 6	18	27	1 ,, 3	12	32	1 ,, 4·5	15	21	1 ,, 3·6	...		...	
4,500	10	26	1 ,, 5·4	...		...	15	57	1 ,, 3·5	...		...	...		...	
5,000	12	26	1 ,, 4·5	...		...	18	27	1 ,, 3	...		...	...		...	

N.B.—Foreign guns and rifles do not differ much from our own in the above particulars. It must be borne in mind that magazine rifles and field howitzers are being adopted rapidly by all nations, and it will be necessary to consider their balliistics when known.

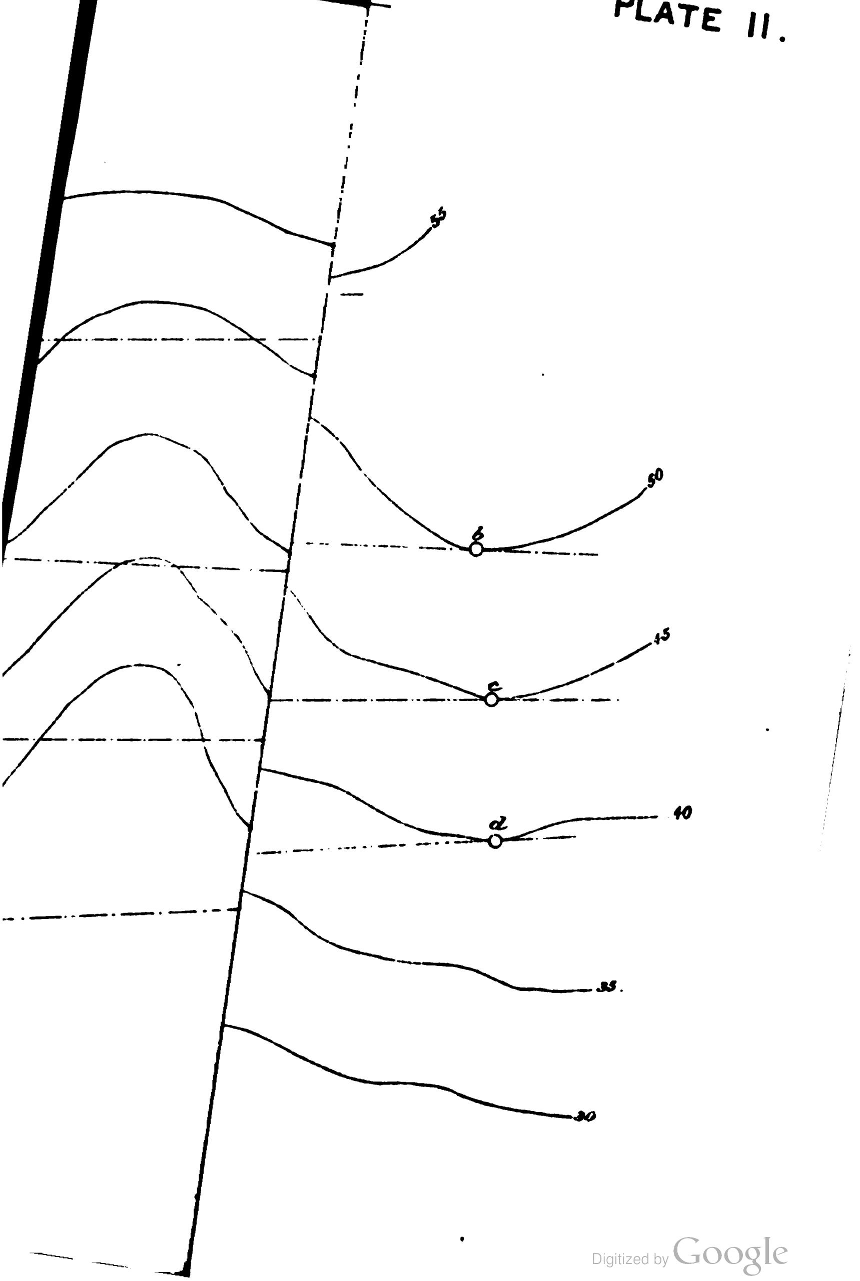
RKS TO IRREG. **PLATE I.**



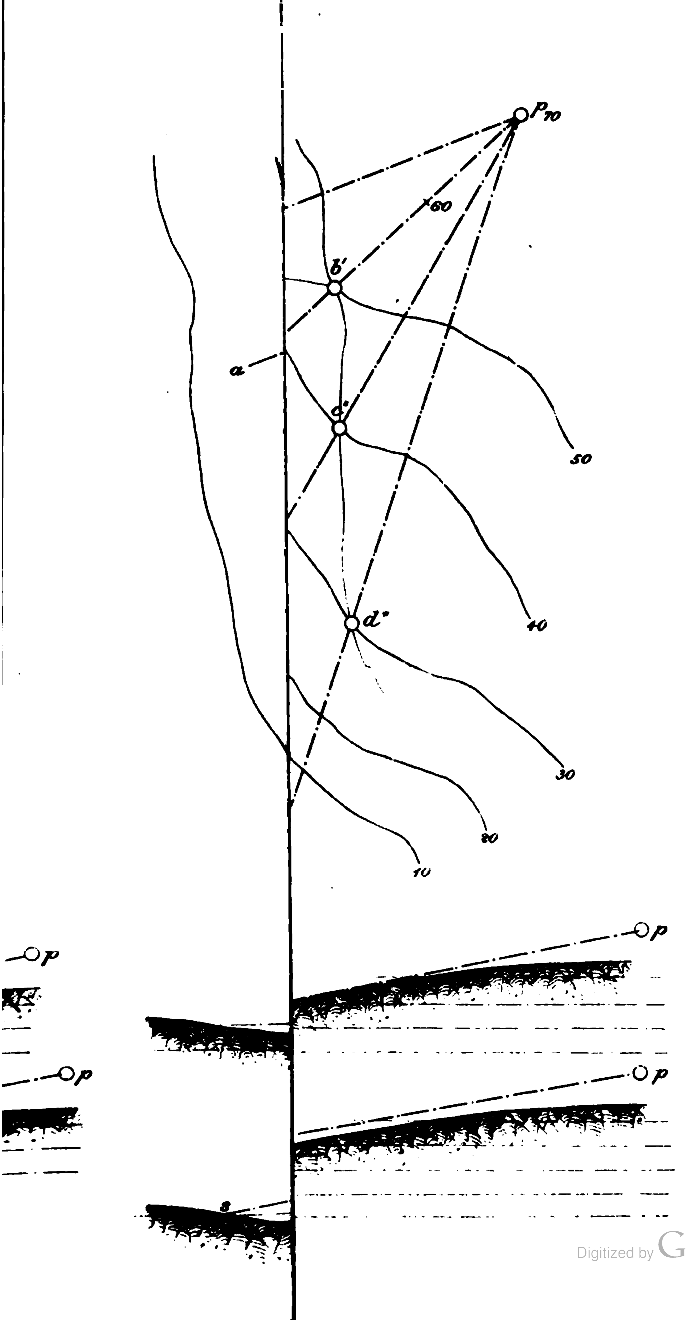


RKS TO IRR .

PLATE II.



F WORKS TO D.   PLATE III.



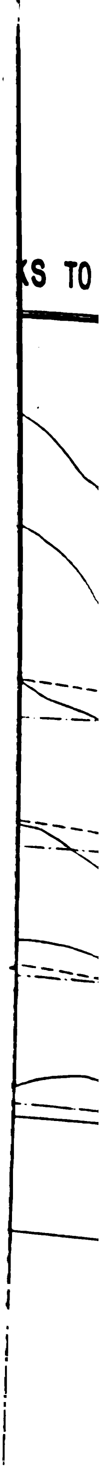

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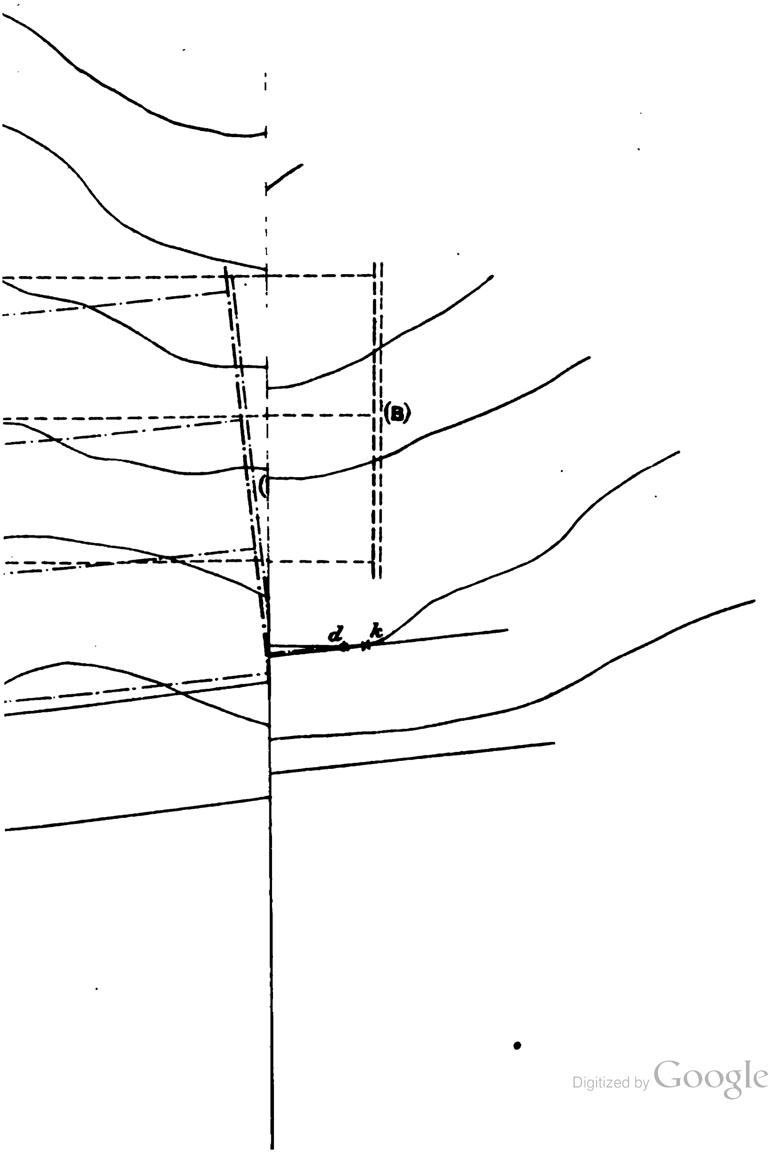
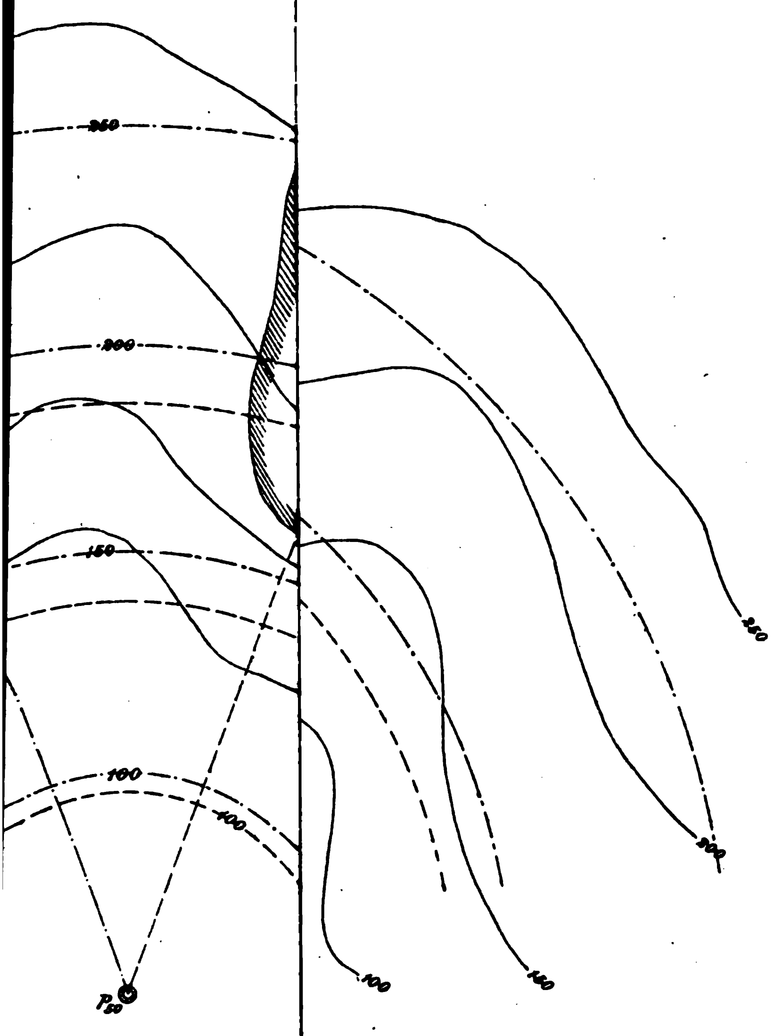
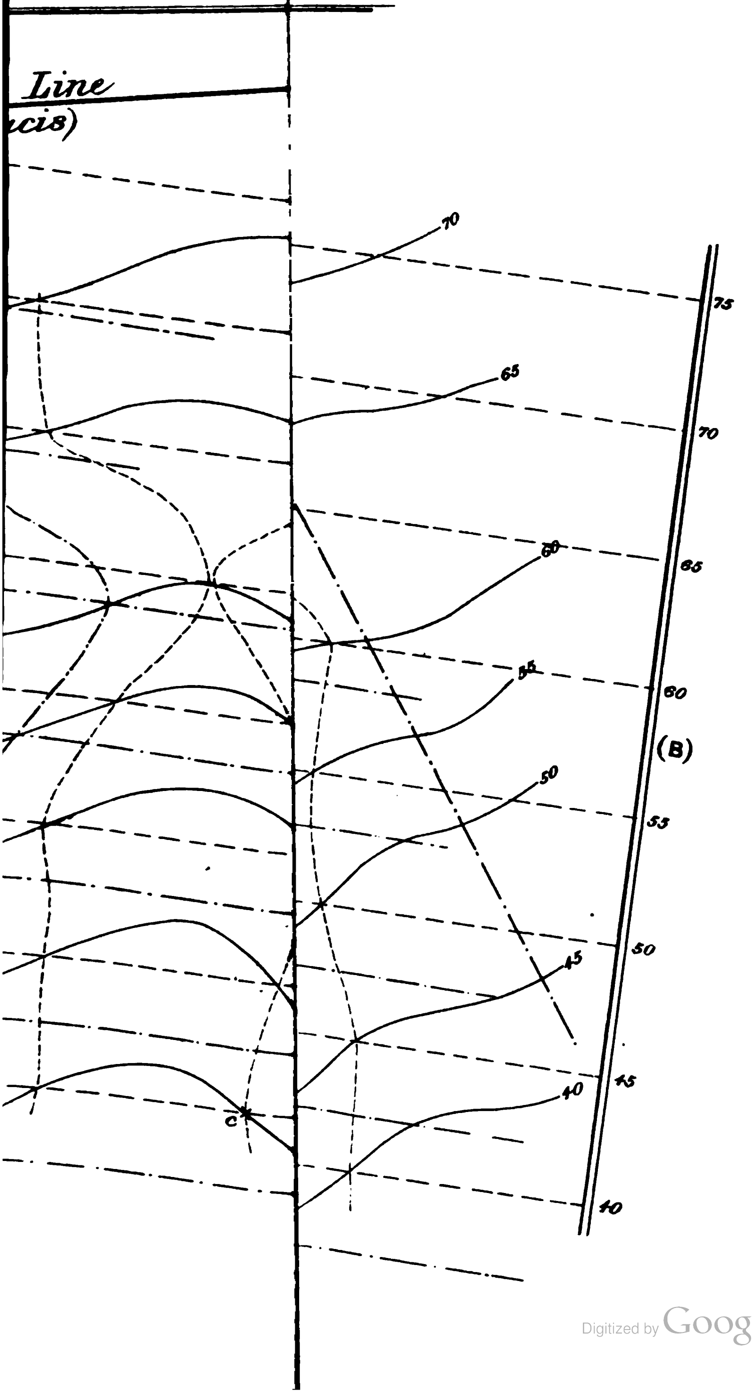
PLATE IV.



RKS TO IRREG. PLATE V.



WORKS TO UND. PLATE VI.



PAPER V.

HYDRAULIC MACHINERY.

By G. R. Bodmer, Assoc. M.I.C.E.

LECTURE I.

It would be impossible for me to deal adequately in the course of three lectures with all that might be included under the title "Hydraulic Machinery," and I shall, therefore, have to limit myself to certain classes only of such machinery. Ordinary pumps, for instance, of which there is a very large variety, would require a course of lectures to themselves, and I shall consequently be obliged to exclude them, giving only a brief description of what are conventionally known as *hydraulic* pumps, this term being applied to pumps working with high pressure and used for giving motion to hydraulic machinery of certain kinds, in contradistinction to pumps employed chiefly for raising water. It is scarcely necessary to remark that a hard-and-fast line of demarcation cannot always be drawn between the two classes.

Weirs and sluices, together with many other kinds of apparatus that might very properly be classified as hydraulic, I must also of necessity pass by.

For practical purposes, the hydraulic machinery of which I propose to treat may, in the first place, be broadly divided into—

(1). Hydraulic lifting and transporting machinery.
(2). Hydraulic tools.
(3). Hydraulic motors.

Under (1) I shall include lifts, cranes, and the pumps and accumulators required to work them.

Under (2), hydraulic rivetters, punching and shearing machines, flanging and forging presses, and similar apparatus.

Under (3), turbines.

Essential Features in Common of (1) *&* (2).—Both classes of machinery (1) and (2) have the essential feature in common that a small force acting on a plunger or piston of small area is used to exert a great force through a plunger or piston of large area, by which some resistance is overcome and work done. The arrangement, with which you are no doubt familiar, is represented in principle by *Fig.* 1, in which a represents the small, and A the large plunger, fitting water-tight in their respective cylinders, which are connected by the passage or pipe B. A force f exerted on a causes to act on A a force F which, neglecting the weight of plungers and fluid and the friction, bears to f the same ratio as the effective area of the plunger A to the effective area of the plunger a. The hydrostatic pressure p per square inch of area is the same on A as on a, if we neglect the difference in the height of the water in the two cylinders (the effect of which is very trifling compared with the pressures generally used), and neglect also the friction which occurs when the fluid is in motion.

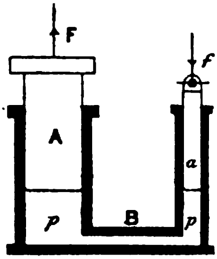


Fig. 1.

The work done on the small plunger by the force f must (in accordance with the law of the conservation of energy) equal the work done by the force F exerted through the larger plunger against an equal resistance.

If, therefore, x denotes the downward motion or stroke of a, and X the corresponding upward motion of A, we have

$$FX = fx ;$$

or, since $F = Ap$, and $f = ap$,

$$AXp = axp;$$

whence
$$\frac{X}{x} = \frac{a}{A} = \frac{f}{F}.$$

I may remark that it is not necessary to assume that p is equal in the two cylinders to arrive at this result, which follows simply from the assumption that water is, for all ordinary practical purposes, incompressible, so that when a certain volume is forced out of one cylinder it occupies that same volume in the other. Instead of *assuming* the equality of pressure, as is so often done, it can be *proved* that this equality must result, under the circumstances, if the principle of the conservation of energy holds good. The fluid may be regarded as a sort of perfectly mobile connecting rod between the two plungers. Mechanically, the action of the two plungers or rams is precisely the same as though they were connected one to either end of a lever having its fulcrum at some intermediate point, the length of the lever-arms being inversely as the areas of the corresponding rams.

By hydraulic transmission, the effects produced by levers can be secured in a much more convenient and compact fashion, there being scarcely any practical limit to the amount of "leverage" or "purchase" obtainable, while the mechanism is very simple.

In practice, the small plunger a is represented by a pump, and between it and the larger plunger a check-valve is inserted in order to enable the pump to do its work by a succession of short strokes instead of one very long stroke. In that case, if x denotes the stroke of the pump, and n the number of strokes required to lift the large plunger, which we will call the *ram*, through the distance X, then

$$nxa = XA.$$

Double-acting Pumps.—For a double-acting pump, having different effective areas during the two strokes, the necessary modification is obvious, as also for several pumps.

Accumulator (*Fig.* 2).— In a large number of instances, the pumps do not work direct into the ram-cylinder, but deliver their water into an accumulator, from which the ram-cylinder is supplied. The accumulator consists essentially of a plunger or piston P, sliding water-tight in a cylinder and weighted sufficiently to exert a pressure per square inch on the effective plunger or piston area equal to the hydraulic pressure which is required. The water is forced into the accumulator-cylinder below the plunger, through the inlet *a* (*Fig.* 2), by the pump, and flows through the outlet *b* into the ram-cylinder; between the pump and the accumulator is a check-valve.

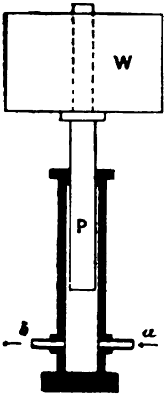


Fig. 2.

If the quantity of water pumped into the accumulator were at all times exactly equal to that flowing out into the ram-cylinder, the plunger P would remain stationary, and the accumulator would then be unnecessary. In the majority of cases, however, the demand for water to supply the ram-cylinder varies considerably, and is often intermittent, while the pumps go on working continuously. It is obvious that when the quantity supplied by the pumps exceeds the quantity leaving the accumulator, the plunger of the latter must rise, while when the demand on the accumulator is greater than the supply, the plunger must fall, but under all circumstances the pressure of the water in the accumulator remains practically the same, since any difference due to variations in the level is quite inappreciable compared with the pressures which are usual in such cases.

Purpose of Accumulator Two-fold.—The accumulator, therefore, fulfils a two-fold purpose: first, it acts as a storage reservoir to equalize fluctuations in demand and supply; secondly, it serves as a safety valve, and prevents the pressure employed from exceeding that due to the load on its plunger.

An accumulator is necessary not only when the requirements of the machine which it serves are *variable*, but also when they are *very constant*, since under the latter conditions the periodical variations in the piston speed of the pumps must not be felt.

A sufficiently large volume of water must always be contained in the accumulator to allow for the maximum difference between the demand and supply without emptying the accumulator-cylinder.

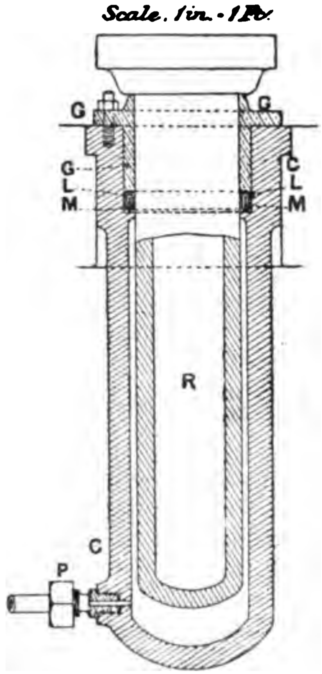


Fig. 3.

Excess of Pressure from Sudden Arrest of Plunger.—When the volume of an accumulator-cylinder is comparatively small, the plunger and the weight supported by it will descend with considerable velocity at the time of maximum demand, and if the outlet is suddenly closed and the descent of the mass arrested, the pressure will rise temporarily much above that due to the load. Where this occurs, allowance must be made for this excess of pressure in calculating the strength of the cylinder walls.

Safety Arrangement.—To prevent any damage being done when the accumulator-plunger has been forced up to the top of its lift, some arrangement is made either for automatically opening a relief-valve, which allows the water set in motion by the pumps to circulate freely without entering the accumulator, or for stopping the pumps.

Before describing the construction of pumps or accumulators as a whole, it will be necessary to say a few words about some of the essential details common to all hydraulic machinery of the kind with which we are now concerned.

These comprise (*vide Fig.* 3): (1), the cylinder C; (2), the ram or plunger R; (3), the packing arrangements for making a water-tight joint between cylinder and ram, consisting of the gland G and leather L; (4), the pipe connections P; and, finally, the valves.

CYLINDER.

Material.—The cylinder is generally either of cast-iron or cast-steel, the latter material having come more and more into use of late years.

Strength.—The cylinder must be strong enough to resist for an indefinite period the maximum working pressure, and the thickness of the walls should, therefore, be calculated for a tensile stress corresponding to the bursting force acting on them. As for the pressure generally used the thickness of the walls becomes very considerable, it is not sufficiently correct, in calculating the strength of the latter, to assume that the stress is equally distributed over the thickness, but Clark's or some similar formula should be used, in which the assumption is made that the stress varies inversely as the radius. The end of the cylinder should be dished sufficiently to insure a stress there not exceeding that in the side walls. For cast-steel cylinders the factor of safety, when Clark's formula is employed, may be from 3 to 3½ for the maximum or test pressure. For cast-iron a higher factor of safety is necessary, as this material has, strictly speaking, no elastic limit, and a slight permanent stretch is produced at each application of the pressure, so that eventually rupture must occur. Cast-steel has about *three times* the tensile strength of cast-iron, and *twice* the transverse strength.

Cast-iron cylinders sometimes burst after working satisfactorily for long periods.

In the case of the cylinders used for the Chignecto hydraulic ship-lift, to which I shall presently refer, the working stress in the innermost fibre by Clark's formula is about 1·75 ton per square inch for a special cast-iron and steel mixture having an ultimate tensile strength of 16 tons per square inch; this is equivalent to a factor of safety somewhat over *nine.* For very high pressures a smaller value than this is usual, otherwise the thickness becomes excessive.

Flaws and Leakage.—It not unfrequently occurs, especially with steel hydraulic cylinders, that leakage through small flaws or blowholes shows itself when the cylinders are tested under pressure. When this leakage is not very considerable, it can often be cured by very simple expedients.

In some cases, by filling the heated cylinder with melted resin, the blow-holes are filled up and cause no further trouble. In other cases, where the leakage takes place through one or two small flaws which can be seen on the surface, these can be closed by hammering, more especially when the material is steel. Very trifling holes sometimes rust up.

Casting Steel Cylinder.—To insure soundness, steel cylinders must be cast with a head at the end which is uppermost in the mould; this head has to be afterwards cut off; it is sometimes formed at one end, sometimes at the other, the practice of manufacturers varying in this respect.

Shape of Cylinder.—It is advisable to make a hydraulic cylinder as plain in form as possible in order to avoid internal stresses.

A stuffing-box for the gland and leather is formed at the open end.

Inlet and outlet orifices are provided near the bottom, which are connected with the pipes conducting the water to and from the cylinder. With suitable valves, one orifice may serve for both inlet and outlet.

THE RAM OR PLUNGER.

The body of the ram or plunger is cylindrical, and is usually of cast-iron, sometimes surrounded with a bush or shell of gun-metal or brass where it works in the cylinder, or, in smaller machines, entirely of gun-metal. At the top it carries a head, to which the special mechanism necessary for any particular purpose is attached.

The ram must be strong enough to resist crushing or collapse, but, as a rule, where cast-iron is the material employed, the thickness of metal, dictated by purely practical considerations, is much in excess of that required to insure the necessary strength. The thickness of the ram is generally rather less than that of the cylinder where both are of the same material. When the diameter is great, it is desirable to calculate the strength of the ram—like that of a boiler flue—for resisting collapse, more especially if constructed of wrought-iron or steel for the sake of lightness.

THE PACKING ARRANGEMENTS.

For the high pressure employed in hydraulic machinery of the type under consideration, the ordinary stuffing-box, with hemp or similar packing, has hitherto generally been considered insufficient, and leather collars and packings of various forms have been resorted to. Recently, however, leathers have in many cases been abandoned by makers of hydraulic machinery, and the same method of packing has been adopted for high pressure of from 700 to 800lbs. per square inch as for what are generally known as low pressures. The reason for this change is to be found chiefly in the trouble often experienced in changing leathers when worn out. Apart from this, leathers are the most efficient method of making a water-tight joint between pistons, or plungers, and cylinders, and it is, therefore, necessary to explain their construction and the manner in which they are used. Leathers are of various forms.

U-leather.—That shown at L in *Fig.* 3 is what is termed a U-leather; it consists of a ring or collar of U-section which fits closely round the ram and lies at the bottom of a recess or stuffing-box at the one end of the cylinder, with its convex surface towards that end. It is kept in place by the gland G, by which, however, it should not be squeezed. To support it in the centre of the U a narrow brass ring M is generally employed resting on the bottom of the stuffing-box. The pressure of the water inside the annular trough formed by the leather keeps one outer surface, that having the smaller diameter, pressed up against the ram, and the other outer surface of larger diameter, against the inside of the stuffing-box.

Hat-leather.—Another form of leather is the "hat"-leather, shown in *Fig.* 4 at H. This is simply an angle ring of leather. One flange is screwed down tight under the gland, the other embraces the ram and is contained in a groove or recess prepared for it on the thickness of the cylinder with sufficient clearance

—which need be very slight—to allow the water to get freely on the outside of it and press it up against the surface of the ram. It will be seen that with this form of leather the tightness of the horizontal joint between the leather and the cylinder depends on the gland being screwed down sufficiently hard.

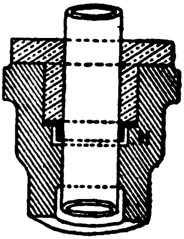


Fig. 4.

Cup-leather.—A third variety of leather, shown in *Fig.* 5, is the cup-leather C, which is used in some hydraulic pumps. It is formed like a cup, with a central hole cut in the bottom, and is screwed or bolted with a large nut and washer to the end of the plunger or ram. The hole allows the screw to pass through for fastening the leather to the ram.

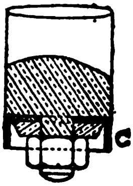


Fig. 5.

With the cup-leather no stuffing-box is really necessary to form a water-tight joint, except when it is used with a piston instead of a plunger for a double-acting pump.

Attempts have been made to substitute some other material for leathers to form hydraulic packing. Metallic collars of forms similar to those of leathers have been tried, but do not appear to have proved satisfactory.

" *Woodite.*"—Recently, however, packings constructed of a new material called " Woodite " are said to have given good results.

Wear of Leathers.—Good leathers will sometimes last for six months, subject to continual wear, before requiring renewal; on the other hand, it will happen that a leather does not wear as many days.

For use with ordinary stuffing-boxes, there are a very large number of packing materials in the market. Those partaking somewhat of the nature of hard indiarubber or guttapercha rings appear to be best suited for hydraulic purposes. In some cases alternate layers of hemp and indiarubber are inserted in the stuffing-box. Generally speaking, for high pressure a harder material than hemp is necessary.

PIPE CONNECTIONS.

Flange Joints.—Where possible, and not too cumbersome, ordinary flanges bolted together are employed to form the joints of high pressure hydraulic pipes. The flanged joint formerly used by the London Hydraulic Power Company is shown in *Fig.* 6, that now adopted in *Fig.* 7. The latter is stated to be 35 per cent. stronger than the former.

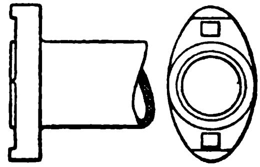


Fig. 6.—*Old Form of Flange.*

The pressure in the mains for which this joint is used is 700lbs. per square inch, and the diameter of the pipes six inches, the material of which they are constructed being cast-iron.

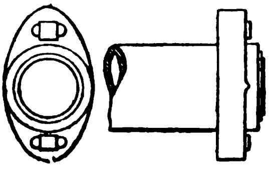


Fig. 7.—*New Form of Flange.*

Nut Unions.—Small pipes up to about three inches diameter are connected by the union illustrated in *Fig.* 8, which represents the attachment of a pipe to the wall of a hydraulic cylinder.

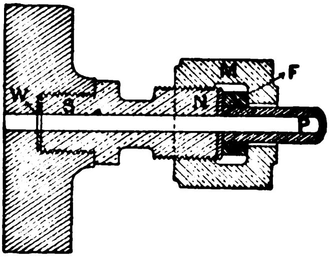


Fig. 8.

The nozzle or nipple N (*Fig.* 8), with a rather fine thread cut on the outside, is screwed by the shank S into the metal of the cylinder wall at the place provided for the inlet or outlet—as the case may be—where a small boss is cast on. A water-tight joint is made between the end of the screwed shank and the surface of the cylinder metal by means of the leather washer W. The pipe P has a narrow flange or collar F screwed and brazed on to its end, and by means of the nut M, which screws over the nozzle N in a manner which is sufficiently clear from the illustration, this collar is pressed up tight against the flat end of the nozzle, a leather washer being used as packing. For joining two pipes, a similar arrangement serves, with the exception that an externally-screwed collar on the end of one of the pipes takes the place of the nozzle N.

Right and Left Screw Union.—Another type of pipe union (*Fig.* 9) is formed by screwing the outside of the pipes for a short distance

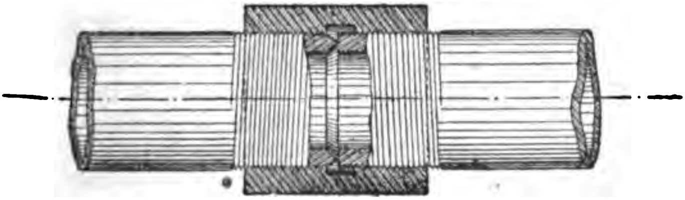


Fig. 9.—*Right and Left-hand Screw Union Joints.*

from the adjacent ends, one with a right-handed and the other with a left-handed thread, and drawing the ends tightly together by means of a nut tapped to correspond. The end of one pipe is made flat, while the abutting end of the other is tapered all round to a sharp edge, in the middle of its thickness, which bites into the flat surface. In many cases a copper washer is interposed between the two pipe ends, but this is not necessary when they are accurately turned.

A pipe connection somewhat similar to that just described is used for the lines of pipe conveying petroleum for long distances across country in America; it was also adopted for the Suakim and Berber water supply.

Right-hand Screw Union.—In this, *both* pipe ends have right-handed threads. The screwed portions of the pipes are slightly coned on the outer circumference, and are connected by a screwed sleeve or nut coned internally to suit the pipes. As the nut is screwed up, it thus becomes more and more tightly jammed on to the pipe ends. The construction is shown in *Fig.* 10.

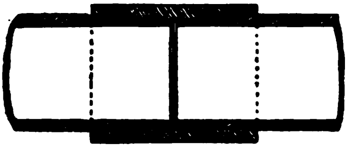


Fig. 10.

A nut is first firmly secured to one end of a pipe length, and the adjoining length is then screwed into it.

Very often the nuts employed with the two preceding joints are cylindrical, and must be screwed up by means of tongs.

For sizes up to about three inches diameter, drawn-steel or wrought-iron pipes suitable for hydraulic purposes are manufactured; beyond this welded pipes become necessary, and these have been made in sizes up to 10 inches inside diameter, for a test pressure of one ton per square inch, and with a thickness of ⅝-inch.

When flanges are used with wrought-iron or steel pipes, they have to be screwed on the outside of the latter and brazed or welded.

Quite recently, pipes manufactured by what is known as the "Mannesman" process have been employed for oil mains.

VALVES.

Balancing Valves.—Generally speaking, especially for small sizes, the valves used in hydraulic machinery with very high pressure are similar in construction to those adopted for low pressures, but when the diameters become large, some means of partially balancing the pressure of water on the valves before they are opened is required. One usual method of effecting this is to furnish the main valve with a small auxiliary valve, which, when opened, admits the water under pressure to both sides of the main valve. As generally constructed, this type of valve can be used for a flow of water through it in one direction only. By an ingenious modification, introduced by Mr. E. B. Ellington, of the London Hydraulic Power Company, and shown in *Fig.* 11, a valve on this principle can be made to act equally well on whichever side the excess of pressure may be, so that the flow can take place in either direction. Assuming, in the first instance, that the valve is closed, with the pressure on the right hand or top of the valve, the water will leak past the leather A through the space B, and by its pressure keep the small auxiliary valve C closed. By giving a few turns to the screwed valve-spindle, the valve C is raised, and the water passes into and fills the pipe D, exerting its pressure on the lower as well as the upper surface of the main valve, and balancing the latter so that it can be opened by continuing to turn the spindle, its weight only having to be raised. If, on the other hand, the excess of pressure, when the valve is closed, is below the latter in pipe D, it

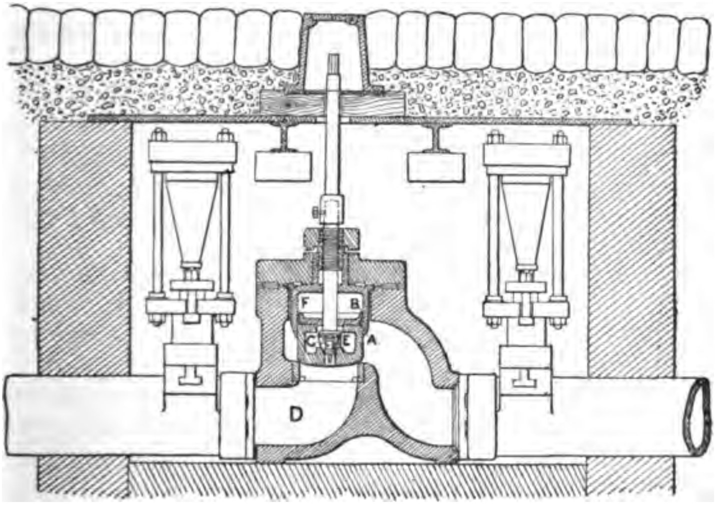


Fig. 11.

keeps the valve C open, pressure is maintained in the cylinder F (through the clearance space between the leather A and the spindle), and the leather is thus kept tight against the sides of the cylinder. The pressure in F on the top of the main valve then balances that in D on the bottom.

It will be seen that an essential feature of this device is that the relief-valve C should have a certain amount of play relatively to the spindle, so that it is capable of opening or closing independently of the latter when in the lower position.

Plunger Valve.—A usual type of hydraulic valve is that illustrated in *Plate* I., *Fig.* 1. This particular valve is designed for the Chignecto hydraulic ship-lift, which I shall presently describe. It is a simple plunger-valve with a spherical seat at the lower end. The pressure on the end is balanced by a weighted lever in a manner similar to that adopted with safety-valves. The valve is opened by means of a hand-wheel and screwed spindle acting on a bell-crank lever. The weighted valve lever is raised by a link from the shorter and horizontal arm of the bell-crank lever; to the end of the long arm of the latter the screwed spindle is connected by a nut swivelling in a fork.

Lüthy's Valve.—For many purposes, a valve known as Lüthy's valve is used. It is a cylindrical slide-valve or piston-valve with U-leather collars or packings, and, as the hydraulic pressure acts equally all round, it is balanced.

To move it, only the friction of the valve and leather has to be overcome. It is successfully employed as an automatically-worked valve making six to seven strikes a minute for certain kinds of presses designed for special work.

Tweddell's Valve.—A modification of Lüthy's valve has been designed and is made use of by Mr. R. H. Tweddell for regulating the admission of water to his rivetting machines. The construction is shown in *Plate* I., *Fig.* 2. The chief improvement consists in the protection of the leathers from the action of the stream of water flowing past them by the annular pieces A1, A2, A3, containing passages through which the fluid is conducted to the admission and exhaust ports without touching the leathers. A series of ports *a*, arranged in the valve all round its circumference, when placed in the proper position, connect through the passages *p* and *c* the inlet P from the accumulator with the outlet C to the hydraulic cylinder, while the recess *e* establishes communication through *c* between C and the exhaust E.

Stop, or Screw-down Valve.—Screw-down or stop-valves for high pressures are constructed as shown in *Plate* I., *Fig.* 3. Sometimes the valve-seat is flat instead of conical, and is furnished with a leather or indiarubber washer to insure a water-tight joint.

Slide Valve.—For hydraulic lifts, the hydraulic cylinders of gun-carriages and other purposes, slide-valves, like that illustrated in *Plate* II., *Fig.* 1, are very general. In principle, they are the same as the slide-valves of steam engines.

Suction and Delivery-valve.—For high pressure hydraulic pumps, the suction and delivery-valves are usually of the ordinary check-valve type, with conical or flat seats sometimes furnished with leather or indiarubber washers.

HYDRAULIC PUMPS AND ACCUMULATORS.

Having now briefly dealt with some of the more important details incidental to all hydraulic pumps, accumulators, presses, lifts, etc., I can now pass on to the general design of pumps and accumulators.

There are so many varieties of these that a few examples only can be here referred to.

ACCUMULATORS.

Plate II., *Fig.* 2, shows in vertical section what is known as a differential accumulator.

Differential Accumulator.—In this case the ram or plunger is stationary, and the cylinder is weighted and slides up and down on it. A is a vertical column, forming the ram, fixed in the base-plate B, and steadied at the top by a wall bracket C. The cylinder D slides upon the column A, and is weighted by the rings E. The lower part of the ram or column A is larger in diameter than the upper part, and it is upon the area due to this difference of diameter that the water acts. Water pumped through the passage F raises the cylinder with its load of cast-iron weights.

Increase of Pressure from Shock.—The indicator diagrams show the temporary increase of pressure due to suddenly arresting the fall of the cylinder with its weight rings. Diagram G was obtained when the accumulator was loaded with 12 weights, each equivalent to a pressure of 100lbs. per square inch in the cylinder, but as the latter itself produces a pressure of 300lbs. per square inch, there is a total static pressure of 1,500lbs., represented by the dotted line H. The area of the diagram above this line represents the temporary increase of pressure due to the momentum of the falling weight. Diagram J was taken with only six weights on the accumulator, while diagram L shows the temporary increase of pressure due to the weight of the cylinder only. I am indebted for these diagrams and particulars to Mr. Ralph Tweddell.

Loading Accumulators.—Very often, instead of cast-iron weights, a large wrought-iron casing is attached by a cross-head to the moving cylinder or plunger of the accumulator, and filled with some heavy material of no value to load it.

Inverted Accumulator.—As before mentioned, either the plunger or piston may be stationary and the cylinder moveable, or *vice versa.* In the latter case the cylinder is sometimes inverted, and the weights hang from the rod of a piston through which the pressure is exerted on the fluid.

Steam Accumulator.—Instead of weights for loading an accumulator, steam pressure acting on a piston which works in a cylinder of suitable diameter, and is attached to a continuation of the accumulator or plunger, has been sometimes employed.

PUMPS.

Armstrong's Pump.—One of the earliest forms of high pressure hydraulic pump is that know as Armstrong's, and illustrated in *Plate* III. It is what is termed a combined piston and plunger pump. All the water is drawn in through the suction-valve S on the forward stroke behind the piston, but the delivery is distributed equally between both the forward and back strokes. During the back stroke all the water taken in on the forward stroke is forced out through the check-valve A, but a portion of it is simply transferred through the passage B to the opposite (or plunger) side of the piston, the remainder only, corresponding in quantity to the difference between the piston and plunger areas, passing out through the delivery-valve C. On the next forward stroke the water previously transferred to the plunger side of the piston is discharged. In this type of pump, cup-leathers are used on the piston, and U-leathers for the plunger. Two cup-leathers, as clearly shown in the illustration, are necessary to keep the piston watertight from opposite sides.

Horizontal Double-plunger Pump.—Another variety of pump is the horizontal double-plunger pump, shown in *Plate* IV., *Fig.* 1. It has two barrels bolted down to a bed-plate; the plungers are co-axial, one working in each barrel. A connecting rod, with a long fork to clear the front barrel, is connected to the central cross-head between the plungers, and is driven through a crank-axle from a pulley on the latter. The adjacent ends of the plungers fit into taper sockets in the cross-head. Only U or "hat"-leathers are necessary in this case.

Diagonal Pumps.—A favourite form of hydraulic pump for small powers is that in which two or three cylinders are arranged diagonally in a tank, from which they draw their water. The tank itself forms the frame supporting the bearings for the crank-shaft, from which the plungers are driven by connecting rods.

Direct-acting Pumps.—Direct-acting pumps—viz., pumps in which the pump piston-rod or plunger is a continuation of the piston-rod of the steam engine by which the pump is driven, there being no crank or fly-wheel—of various design are applicable to high-pressure work.

Among these is the well-known "Worthington" pump, which is really a development of the "Duplex" pump, introduced more than 12 years ago into this country from America. It consists essentially of two direct-acting pumps fixed side by side, parallel with each other; its leading characteristic is that the slide-valve belonging to one of the steam cylinders is worked from the piston-rod of the other. The relative position of the reciprocating parts is such that when one piston is at the commencement of its stroke the other is at the middle. Such pumps can be started in any position of the pistons, and can be driven either quickly or slowly, if necessary, so slowly that the motion can hardly be perceived at a cursory glance.

The "Duplex" pump, as originally constructed in this country, is shown in *Plate* IV., *Figs.* 2, 3 and 4; the arrangement of the combined piston and plunger and valves is that of the "Armstrong" pump, and has already been referred to in connection with the latter.

Drawbacks of Direct-acting Pumps.—The chief drawback to direct-acting steam pumps is, generally speaking, their wastefulness as regards steam consumption. This arises from the fact that the steam cannot, with the usual construction, be worked expansively to any considerable extent, because the inequalities between the resistance of the pump and the force acting on the steam piston would otherwise become too great, and cause inadmissible variations in the speed. In pumps which are connected with a crank, the differences between the total steam pressure and the load can be compensated by means of a fly-wheel, but with direct-acting pumps of the kind in question, fly-wheels cannot be used.

"*Worthington*" *Pumps.*—In the latest form of "Worthington" pump, this difficulty has been overcome, partly by "compounding" the steam cylinders, and partly by the use of a device, which I shall shortly describe, for storing up the excess of energy developed by the motor during the first half of its stroke, and giving it out again during the last half when there is a deficiency.

Probably the largest and most powerful hydraulic pump yet made is the "Worthington" pump shown in *Plate* V., *Fig.* 1. It consists of two twin steam pumps side by side; each pump has two plungers of 12 inches diameter working through stuffing-boxes at the outer ends of the pump barrels, which have the same centre line and are connected in the middle. The plungers are connected by tie-rods, and are worked direct from the continuation of the piston-rod of the high-pressure cylinder. The steam engine is of the compound tandem type, with cylinders 41 and 82 inches in diameter. The stroke of both steam pistons and plungers is about 40 inches, but varies somewhat.

The hydraulic pressure in the mains is about 1,500lbs. per square inch, the steam boiler pressure 100lbs. per square inch, and the engine exerts about 800 horse-power. The pump is used in America for forcing petroleum through long pipe-lines.

Compensating Cylinders.—The device previously alluded to as fulfilling the functions of a fly-wheel in this class of pump, and known as a "compensator," is constructed and works as follows:—To the end of the tail plunger are pivotted two connecting rods attached to pistons working in oscillating cylinders fixed symmetrically above and below the centre line of the pump. These compensating cylinders are filled with fluid, and communicate with an air-vessel under pressure from the main. During the first half of each stroke the air in the air-vessel is compressed by the pistons, and, in effecting the compression, the excess of work developed in the steam cylinders over that required for the pumps is absorbed and momentarily stored in the air-vessel. During the last half of the same stroke the compressed air expands and restores the previously absorbed work to the plungers. This follows from the arrangement of the compensating cylinders, which are so placed that about the middle of the stroke of the pump their axis is perpendicular to the pump axis.

During the first part of a pump stroke—as will be easily understood from the illustration—the pistons of the compensating cylinders are being pushed inwards, while during the latter part of a pump stroke, after the perpendicular position has been attained, the same pistons move outwards.

Steam Valves.—In compound Worthington pumping engines of the type in question, cut-off valves are used with the high-pressure steam cylinder. The main distribution-valve is an ordinary slide-valve, and is worked from the piston-rod of the fellow engine on the "Duplex" principle. The cut-off valves are cylindrical valves oscillating in separate chambers, and worked direct from the piston-rod of the engine to which they belong.

Curves Showing Compensating Actions.—The curves shown in *Plate* V., *Fig.* 2, represent the distribution of power throughout the stroke in the high and low-pressure steam cylinders of a large Worthington pumping engine working at Hampton, the energy absorbed and restored by the compensators, and finally the resultant distribution obtained by the use of the latter.

The lowest line CF is the curve due to the compensators themselves. The area CDY represents the work absorbed during the first half of the stroke, and the area FEY the energy restored during the second half of the stroke. To arrive at the resultant curve for the distribution of the work, the ordinates for the first part DY of the stroke have to be deducted from those of the steam-power curve, and the ordinates of the second part YE added to those of the steam-power curve at the corresponding points. It will be seen that the result is fairly constant aggregate pressure throughout the stroke.

Worthington pumps are now being used with *triple* expansion engines, and for these "compensators" are found to be unnecessary.

SPEED OF HYDRAULIC PUMPS.

The speed of a pump is limited chiefly by the rapidity with which the valves can be made to perform their duty satisfactorily, and in practice it is not usual to exceed 60 strokes a minute—double or single, as the case may be—although there are pumps running at 100 strokes. In a pump with valves worked mechanically, like those of a steam engine, a higher speed is possible, especially if the water enters the pump under a slight head.

Losses in Hydraulic Machinery.

In no other class of machinery are the losses, under favourable conditions, so small, and consequently the efficiency so high, as in hydraulic machinery in which great pressures are employed. The reason for this is not far to seek.

The power exerted by a pump, for instance, is directly proportional to the pressure used, while the loss from friction of the mechanism and of the water flowing through the passages and pipes does not increase in the same ratio.

Pipe Friction.—It has been found that the friction of water in pipes is practically independent of the pressure, and depends for given dimensions only on the velocity of the flow, so that obviously, as far as this is concerned, it is advantageous to use as high pressure as possible. The usual formula, as you are probably aware, for the loss of head by friction of water flowing through cylindrical pipes is $h = \zeta \frac{l}{d} \frac{v^2}{2g}$, where h is the loss of head, l the length of pipe, d the diameter of the same, ζ an experimental co-efficient, and v the velocity of flow in feet per second of the fluid. The co-efficient ζ is not constant, but varies with the velocity, decreasing as the velocity increases.

Hawksley's well-known formula for the number of gallons of water which can be delivered per hour through a pipe of given diameter and length with a given head is only another form of the same equation, in which the velocity of the water is expressed in terms of the quantity flowing through the pipe in a given time and the diameter of the pipe.

The loss of head incurred in forcing a certain quantity of water through a pipe in a given time is, according to Hawksley's formula, *inversely* proportioned to the fifth power of the diameter.

Material of Pipes Unimportant.—As you are no doubt aware, the friction is nearly independent of the material of which the pipes are composed, at any rate with the materials used in practice.

Losses in Bends, etc.—Other sources of loss are the passage of the water through pipe-bends and knees; these, therefore, should be avoided as much as possible in designing hydraulic machinery.

Loss by Shock.—Loss is frequently incurred by shock due to sudden change of velocity in pipes or passages.

The formulæ and co-efficients for calculating such losses are to be found in the ordinary text-books on hydraulics.

Friction of Leathers.—When leathers are used for packing, the friction of pistons and plungers is very small.

Some years ago experiments were carried out by Mr. John Hick and the late Mr. Lüthy with the object of determining the relative amount of this friction. The results demonstrated that the friction F increases with the diameter d and pressure p, but is independent of the depth of the leather. In percentage of the total pressure on the ram or plunger, the friction was found to be

0·1 per cent. for a ram of 4 inches diameter.
0·25 " " 8 " "
0·5 " " 16 " "

It can be expressed by the formula $F = 0{\cdot}0471dp$ for new leather or bad lubrication, and $F = 0{\cdot}0314dp$ for leather in good condition.

Velocity of Flow in Pipes.—A question intimately associated with the subject of pipe friction is that as to the admissible velocity of flow of the water in pipes under given circumstances.

The velocity is limited in the first place by considerations of economy in power, and secondly by the necessity for avoiding excessive strains in the material of the pipes which would result from the sudden stoppage of a large mass of water in rapid motion.

As the loss by friction is independent of the pressure, obviously the *percentage* of loss, due to a certain velocity of flow in a pipe of given dimensions, is *less* the *greater* the pressure (or available head); hence, on economical grounds, a higher velocity is allowable with a great pressure than with a low pressure.

With regard to safety, the speed of the water in a short pipe may exceed that in a long one.

In the 6-inch mains of the London Hydraulic Power Company, with a maximum length of three miles, the velocity of flow with all the sets of engines and pumps working is about 2·83 feet per second.

In the 6-inch pipes supplying water to the Chignecto ship-lift, the velocity is over 11 feet per second.

The pressures in the two cases are not very different, but the mains for the Chignecto lifts are comparatively short.

In large American oil pipe-lines, with a diameter of six inches, the velocity is about 4¼ feet per second with a length of 20 miles. The Admiralty allow a maximum velocity of 10 feet per second.

APPLICATION OF HYDRAULIC POWER.

I now propose to give you an account of a few of the many purposes to which hydraulic power, acting through rams or pistons, is applied, and shall endeavour to select those which are of most practical importance.

In all cases a hydraulic pump or set of pumps is required, often in conjunction with an accumulator. Where several machines are driven, the pumps or accumulators deliver the water into a main, from which branches lead to the various machines. Every such installation is an example of the hydraulic transmission of power, but within the limits of a machine-shop, manufactory, or warehouse, the loss in transmission is so slight as to be nearly negligeable, unless the branches are very numerous and a large number of machines are in use simultaneously. An example of the hydraulic transmission of power on a large scale is afforded by the system established by the London Hydraulic Power Company. The power transmitted is used for working cranes and lifts, and also for driving hydraulic motors.

In accordance with the plan laid down at the commencement of my lecture, I shall begin with

HYDRAULIC LIFTING AND TRANSPORTING MACHINERY.

This comprises cranes and travellers, capstans, jacks, and lifts for passengers, goods and ships. I shall, of course, not attempt to deal exhaustively with all these, but will describe and illustrate a few typical examples, from which you will be able to form a fairly complete idea of the character of such machinery.

The credit of introducing on a large scale the class of hydraulic appliances which I am about to describe belongs to Sir William Armstrong.

HYDRAULIC CRANES.

Hydraulic power is applied in various ways to the working of cranes and travellers. In some cases it is used only for the hoisting motion, in others for hoisting, traversing, and turning or slewing.

Armstrong's Hydraulic Jigger.—Armstrong's so-called hydraulic "jigger" is one of the oldest and best-known devices for actuating cranes by means of hydraulic power. The jigger consists essentially of a hydraulic plunger, working in a stationary cylinder, and carrying in forked bearings at its end one or more chain-pulleys or sheaves, over which passes a chain fixed at one end to some part of the cylinder, and at the other end connected with the crane mechanism. It is shown in *Plate* V., *Fig.* 3, and *Plate* VI., applied to wall cranes, but in an improved form for the hoisting motion, with two plungers contained in the same cylinder, one for raising light, and the other heavy, loads.

In this instance the hydraulic power is used both for hoisting and slewing. The cylinder of the jigger for hoisting is placed vertically at the back of the wall. The chain is attached to the bracket at the lower end of the cylinder, passes alternately over the sheaves carried at the end of the plunger, and over those revolving in stationary bearings in the bracket, and thence to a guide-pulley at the end of the jib.

In order to obtain a different amount of power, according to the load to be raised, two rams are employed, the larger of which is hollow, and forms the cylinder for the smaller ram. When the larger plunger is required, it is temporarily connected at its upper end with the smaller plunger by means of two hooks or catches.

It will be seen that the vertical forces acting on the jigger are balanced, with the exception of the weight, in such a manner that no stress is produced in the holding-down bolts or the brickwork. The horizontal pull on the chain gives rise only to crushing stresses.

For the slewing motion there are two smaller horizontal jiggers, each connected with one end of a single chain, which passes round a pulley on the crane-post, and causes the latter to rotate.

The crane illustrated in *Plate* V., *Fig.* 3, is shown fitted with a "Priestman" dredger. The lift is 80 feet, and the (maximum) load 25cwt. The speed of working is such that three complete lifts are made in two and a-half minutes, including filling, lifting, swinging, discharging, re-swinging and lowering the bucket. The water is supplied from the mains of the Hydraulic Power Company at a pressure of about 700lbs. per square inch.

It is obvious that each pair of sheaves—one attached to the plunger and the other to the cylinder—multiplies the stroke or travel of the ram by two; thus with two pairs of pulleys the lift of the chain is four times the stroke of the plunger, while, of course, the forces exerted are in the inverse proportion.

Another type of jib-crane, also actuated by hydraulic jiggers, is shown in *Plate* VII., designed for the Royal Arsenal at Woolwich. It is intended for a load of three tons, to be worked by a hydraulic pressure of 700lbs. per square inch, the test pressure being 2,500lbs. per square inch.

The jiggers are essentially the same in construction as those for the wall-crane previously described. For the hoisting motion the jigger has two plungers, with diameters of 7 and 10 inches.

For the slewing motion there are two horizontal jiggers, with plungers four inches diameter, bolted to the bed-plate carrying the bearing in which the vertical crane-post swivels.

The crane-post is built up of wrought-iron plate and angle-irons, and carries the hoisting jigger inside it so that both turn together.

In consequence of this, the 1¼-inch steel pipe, through which water is supplied to the jigger cylinder, has its axis coinciding with the axis of rotation of the crane-post and swivels in a stuffing-box attached to a casting, uniting it with the fixed supply pipe.

At its lower end, the crane-post has a pivot on which it turns in a bearing, and is furnished with a pulley by which the slewing motion is transmitted from the chain, actuated by the horizontal jiggers. At about half its height, the crane-post is supported in a second bearing, forming the top of a wrought-iron casing bolted to the foundation. This casing has to resist the overturning and bending movement resulting from pressure on the bearing. In the bearing the crane-post is surrounded by a circular cast-iron disc which runs on anti-friction rollers.

"*Bessemer*" *Crane.*—For foundry work, and more especially in connection with steel furnaces, a class of hydraulic crane of the type known as the "Bessemer" is made use of.

In this, the jib, which is rigid and horizontal, is raised bodily by a hydraulic plunger, which itself forms the crane-post, or more frequently the plunger is stationary while the hydraulic cylinder to which the jib is attached moves up and down.

The chain carrying the load is suspended from a "crab" or carriage running on the top of the horizontal girder or girders forming the jib. The crab is made to travel in or out by means of chains, which can be worked by a jigger. The slewing motion may also be actuated by a jigger. In some cases, however, small hydraulic reciprocating motors are employed to produce both the travelling or "racking" motion of the crab and the slewing motion.

The drawback to the ordinary construction of Bessemer crane is that the plunger, being at the same time the crane-post, is subject to a bending strain, and has to be made of much larger diameter than would otherwise be necessary with water under high pressure, in order to resist this strain.

To avoid this evil, the plan has been recently introduced of enclosing the hydraulic cylinder and plunger in an independent frame crane-post of suitable strength, which takes the whole of the thrust and pull exerted by the jib and its tie-rods, and the bending strains caused by them. The plunger has then nothing to do but overcome the weight and friction due to the jib and load; it is subject only to compressive stress.

Plate VIII., *Fig.* 1, shows this system applied to a steel foundry crane. The wrought-iron frame E constitutes the crane-post and swivels in a bracket at each end. To the frame E, inside it, is bolted the hydraulic cylinder J, in which works the lifting plunger or ram A. To the inner end of the jib are attached two rollers D, which run on the face of the crane-post and transmit to it the thrust of the jib. At the inner end of the tie-bars are rollers C, running on the back of the crane-post, which transmit to the latter the pull of the tie-bars. The end of the jib is connected with the hydraulic plunger, and the bearings for the rollers C and D are also connected with each other by the piece B.

The modifications necessary for a free crane-post are obvious; in that case the jib can, if required, be prolonged to the opposite side of the post and provided with a balance weight.

For serving steel furnaces, Bessemer cranes are frequently so made that a foundry ladle is supported by trunnions in bearings at the end of the jib.

HYDRAULIC LIFTS.

Two Systems.—One of the most general applications of hydraulic power is to the working of lifts both for goods and passengers. Broadly speaking, this is effected by two methods analogous to those which I have described in connection with cranes. One method consists in the use of hydraulic jiggers, by which a cage suspended from chains is raised and lowered; the other in attaching the cage or platform directly to the top of a hydraulic ram, with which it rises and falls.

Jigger Lifts.—Plate VIII., *Fig.* 2, represents an example of the first method. The jigger ram working in the cylinder A is inverted, and its weight partly balances the weight of the cage B. The cage is suspended from two wire ropes R, each of which, as a matter of safety, is strong enough to carry the weight. These ropes are led over pulleys and the opposite ends attached to counter-weights W (which, together with the weight of the ram, approximately balance the load).

The chain C is fixed at one end to the cylinder A, and at the other to the counter-weights W.

When the jigger ram is moved downwards by the pressure of water in the cylinder, the cage is raised in exactly the same way as the load on one of the cranes previously referred to, the stroke of the ram being multiplied in proportion to the number of pairs of sheaves used on the jigger.

Another arrangement of jigger lift, in which the ram is horizontal, may be seen in *Plate* VIII., *Fig.* 3.

The lifting chain is sometimes balanced by suspending from beneath the cage a loose chain, which lies coiled up on the ground when the cage is at the bottom and is picked up as it ascends.

The valves of hydraulic lifts are usually controlled from an endless rope, vertically arranged so that the attendant can grasp it in any position of the cage, and passing over four guide-pulleys. This rope is wound several times round a drum connected with a screwed spindle, as shown in *Plate* VIII., *Fig.* 3, at V in the general drawing, and in detail to a larger scale in *Plate* VIII., *Fig.* 4. The screwed-spindle works in a nut fixed on the end of the valve-spindle, and when the drum is rotated, the valve, which is of the slide-valve type, is moved in one direction or the other, and opens and closes the admission and exhaust ports. In the example shown the drum is about 12 inches in diameter.

Direct-acting Hydraulic Lift.—The type of lift in which the cage is attached to the top of a hydraulic ram, known as the "direct-acting" lift, is shown in *Plate* IX., *Fig.* 1, with details in *Figs.* 2 and 3 of the same plate.

The cage is supported on a cross-head or platform, in the lower side of which the top end of the ram is centrally fixed. The stroke of the ram is in this case equal to the total lift, as no multiplying gear is used, consequently the length of the ram and cylinder is very great, and a bore-hole has to be sunk in the ground, to a depth somewhat in excess of the height of the lift, to contain the cylinder.

The weight of the lifting ram and cage is balanced by counter-weights suspended from chains attached to the cage or ram-head at one end, and passing over pulleys supported on beams above the lift. It is obvious that as the ram rises the pressure of the water on its lower end is diminished by an amount corresponding to the rise of the water in the cylinder. With low pressures this diminution may be relatively very considerable, but it can be compensated for by the weight of the chains.

Calculations Relating to Balance.—If G denote the weight of the cage; R the weight of the ram; c the weight of the chain for a length equal to the lift; W the weight of the counter-weight; h the length of the chain corresponding to the weight c, or the height of the lift; x the height through which the cage has been raised at any point of the stroke; w the weight of water corresponding to the displacement of the ram; F the frictional resistance of the mechanism; then for any position of the ram when at rest we have

$$G + R \mp F + \frac{h - 2x}{h} c = W.$$

When the cage is at the top of the lift

$$x = h;$$

hence $$G + R \mp F - c = W.$$

When the cage is at the bottom of the lift

$$x = o;$$

hence $$G + R \mp F + c = W.$$

This is on the assumption that no hydraulic pressure is being exerted on the ram.

The − sign gives the minimum value of W for security against downward motion, the + sign the maximum for security against upward motion. This allows a certain margin, due to the friction, within which W may vary without overbalancing resulting. For the two extreme positions it will be found that the maximum value of W admissible when the cage is at the top of the lift will satisfy the necessary conditions when the cage is at the bottom of the lift, only provided that c does not exceed F. This is, of course, not usually the case, so that in general a *perfect* balance is not possible. As a matter of fact, a certain amount of unbalanced weight is necessary to cause the cage and ram to descend when empty.

When the cage is moving upward, and, therefore, pressure is being exerted on the ram, if P denote the total pressure on the effective ram area due to the hydraulic pressure above the atmosphere when this is a maximum and the cage at the bottom of the lift, then if L is the net load raised

$$\mathrm{L}+\mathrm{R}+\mathrm{G}+\mathrm{F}=\mathrm{W}-\frac{h-2x}{h}c+\mathrm{P}-w.$$

Now w, the weight of water displaced, is proportional to the stroke of the ram x, and, if A denote the effective ram area,

$$w=s\mathrm{A}x,$$

where s is the weight of the unit volume of water. Hence

$$\mathrm{L}+\mathrm{R}+\mathrm{G}+\mathrm{F}=\mathrm{W}+\mathrm{P}-\frac{h-2x}{h}c-s\mathrm{A}x\,;$$

or otherwise expressed

$$\mathrm{L}+\mathrm{R}+\mathrm{G}+\mathrm{F}=\mathrm{W}+\mathrm{P}-c+\left(\frac{2x}{h}c-s\mathrm{A}x\right).$$

In order that there may be equilibrium in all positions of the cage, the value of the quantity within the brackets must be *nil*, that is

$$\frac{2x}{h}c=s\mathrm{A}x\,;$$

or

$$\frac{2c}{h}=s\mathrm{A}\,;$$

whence

$$\frac{c}{h}=\frac{s\mathrm{A}}{2}.$$

Now $\frac{c}{h}$ is the weight of a unit length—say one foot of the chain (or combined chains where several parallel chains are employed)—and $sx\mathrm{A}$ is the weight of a unit length of a column of water of a cross-section equal to the effective ram area.

Rule for Weight of Chain.—To balance the loss of head arising from the increasing height of water in the cylinder, the weight of a foot of the chain—or combined chains—must be equal to half the weight of a column of water one foot long and of a cross-section equal to the ram area.

This is not difficult to see without any calculation.

It is only necessary to take the loss of head in question into account with low pressure.

Safety of Direct-acting Lift. — One great advantage of a direct-acting lift is its comparative *safety.* The cage cannot descend at a greater speed than corresponds with the escape of the water through the outlet valve.

The safety, as has been pointed out by Mr. E. B. Ellington, in a paper read before the Institution of Mechanical Engineers in 1882, is not so great as might at first sight be supposed, when balance weights and chains are employed. The upper portion of the ram, for a considerable length, instead of supporting the cage as a column, is suspended from it, and is, therefore, in tension. If, then, the ram were to break anywhere within this part, or the attachment between the ram and cage gave way, the cage would be drawn violently to the top by the counter-weight. This actually happened at the Grand Hotel in Paris.

Hydraulic Balance.—To prevent the possibility of such accidents, various forms of so-called hydraulic balances have been devised, more or less on the principle originally proposed by Messrs. Tommasi and Heurtvise.

In *Plate* IX., *Figs.* 4 and 5, one construction of such a balance is shown, which will serve to make the principle of action clear.

A hollow ram R (*Fig.* 5) works in the cylinders C and G, and through a stuffing-box in the upper end of the ram passes a pipe B. The space J of the lower cylinder communicates through the aperture H with the lift ram; the annular space EE can be connected through a valve with either the pressure main or the exhaust; while the pipe B is always open to the pressure main. The working pressure acting on the area of the central pipe B constantly balances within a small amount the minimum weight of the lift-ram and cage. When admitted to the annular space EE, the pressure on the area of the ram R outside the pipe B is sufficient to overcome friction and raise the net load. With pressure, therefore, in both spaces B and E, the water in J is forced through the aperture H into the lifting-cylinder and raises the lifting-ram and cage. When the top of the lift is reached, the valve closes the admission port to EE and the lift stops. For descending, the valve opens the exhaust port and the space EE is relieved from pressure. The weight of the ram and cage on the water in the lift-cylinder transmits the pressure through J to the ram R and overcomes the weight of the latter, and the pressure on the area B, forcing the water back through the pipe B into the pressure main and accumulator with which it is connected. The ram R is thus always approximately in equilibrium with the lift-ram. The work and water expended on raising the weight of the lift-ram and cage when the lift ascends is restored—very nearly—when it descends.

Pressure in Lift may be Lower than in Main.—One advantage of a balance of this construction is that the pressure in the lift-cylinder may be lower—to any convenient extent—than the pressure in the mains. This enables the lift-ram to be made of a diameter sufficiently great to insure the requisite stiffness as a column.

If the full pressure in the main were used in the lift-cylinder, the diameter of the ram necessary to raise the load would in some cases be so small that there would be a risk of the ram buckling under the weight.

Combined Hydraulic and Weight Balance.—The hydraulic balance pure and simple, as just described, has certain drawbacks, which have led to the introduction of a lift-balance (*Plate* IX., *Fig.* 6), in which the weight of the ram and cage is neutralized by weights, acting on a combined moving cylinder and ram, transmitting their pressure through the water-column in the cylinder B to the lift-ram. The water from the main for working the lift is admitted through the pipe C, and its pressure acts on the area corresponding to the outer diameter of C. It will be seen that the weights take the place of the water pressure on the pipe area in B in *Fig.* 5.

The drawbacks previously referred to in connection with hydraulic balances of the older type arise from the fact that water for the balance is withdrawn from the mains and forced back again into them whenever the lift rises and falls. It may happen that this is going on simultaneously with several lifts in close proximity, and results in an excessive local pressure on the mains, while in addition, water is temporarily withdrawn from the mains and *not paid for* at a time when it may very possibly be wanted elsewhere.

The balance illustrated in *Fig.* 6 has also the advantage of requiring one water-tight stuffing-box less than the older form.

THE "EDOUX" LIFT.

The Edoux lift at the Eiffel Tower is an ingenious combination of the direct and indirect-acting type.

One of the cabins (No. 1) is supported direct on two rams, each of

124 square inches area, while the other cabin (No. 2) is suspended by four steel-wire ropes passing over pulleys at the top of the lift, and attached at their opposite ends to cabin No. 1; the weights of the two cabins thus balance each other.

The lift of the ram is 263 feet, probably the greatest direct lift in existence. By using the two cabins connected in the manner described, a total lift of 526 feet can be effected, divided into two stages, the passengers being transferred at the end of the first stage from cabin No. 1 to cabin No. 2.

LECTURE II.

THE "OTIS" LIFT.

One of the most recent and best-known developments of the indirect-acting type of hydraulic lift is the "Otis" lift. Two of these machines on a large scale, and specially designed for their purpose, are in use in the Eiffel Tower for raising passengers from the ground floor to the second platform at a height of 380 feet.

General Description.—The Otis lift works with comparatively low pressure, the water being supplied as a rule from a tank on the roof of the building in which the lift is placed. The usual arrangement is vertical, as shown in the general perspective view (*Plate* X., *Fig.* 1).

The cage is suspended from wire ropes passing over a pulley supported on the framework above the lift. These ropes then pass under sheaves, attached to the end of a double piston-rod, connected to a piston working in a hydraulic cylinder, and the opposite extremities of the ropes are attached by shackles to the framework supporting the pulley. The cage is raised by admitting water under pressure *above* the piston and exhausting below; while it is lowered by placing both ends of the cylinder in communication with each other. In the latter case, the weight of the cage draws up the piston and transfers the water above the piston to the space below.

Diagram of Valves.—The valve controlling the admission and discharge of water is in principle a slide-valve of the piston type.

Fig. 2, *Plate* X., indicates the various positions of the valve: (A), when the cage is ascending; (B), when it is descending; and (C), when it is at rest. In this diagram the valve is represented as an ordinary slide-valve.

In position (A), water enters through the supply pipe (1) and passes through the port (2) to the space above the piston F, while the fluid below the piston escapes through the exhaust ports (3) and (4), and off-flow pipe J. In position (B), ports (2) and (3) are placed in communication with each other, while they are shut off from ports (1) and (4). In position (C), ports (2) and (3) are entirely cut off from each other and from ports (1) and (4), so that the apparatus is locked.

"Otis" Lift in Eiffel Tower.—In the Eiffel Tower the cage or cabin of each lift, instead of moving vertically, runs on very steep inclines, and the lift-cylinder is placed at a corresponding angle with the horizon. The general arrangement is explained by *Plate* XI., *Fig.* 6. *Plates* XI. and XII., refer to the Otis elevator as constructed for the Eiffel Tower.

The hydraulic cylinder H (*Fig.* 1) has a diameter of 38 inches and a length of 36 feet. The piston is connected by two rods, 4¼ inches diameter, which are attached direct to a pulley-truck, on which are mounted the grooved pulleys Y, six in number (*Figs.* 1 and 6).

Arrangement of Cylinder.—The cylinder is supported on two girders about 121 feet long, inclined at an angle of 61° 20′. On these girders runs the pulley truck, which travels on wheels.

Tackle.—At the upper end of the girders are six stationary pulleys corresponding with the moveable pulleys Y, the whole thus constituting with the rope a gigantic inverted 12-purchase tackle.

Ropes.—The rope is of steel wire, and quadruple, being composed of four ropes of 0·79-inch diameter. One end (*vide Fig.* 7) is secured to the top of the girders by means of a whipple-tree, to insure equal tension in each of the component ropes, and the latter then pass in succession over the fast and movable pulleys, and are led over guide-pulleys above the second platform of the tower. There the four ropes are divided into pairs, passing down each side of the guides, and are attached with the interposition of a safety apparatus to the cage or cabin.

Counter-weight.—The dead weight is partially balanced by a counter-weight (*Fig.* 7), a sufficient weight being left unbalanced to enable the cage to descend of itself when empty and lift the pulley truck and the piston. The counter-weight consists of a truck 27 feet long, running on four wheels, and loaded with cast-iron weights. It travels on a track 148 feet long laid on girders, beneath the main cage track near the lower end, and inclined at an angle of 54° 35′.

Ropes for Counter-weight.—The two steel-wire ropes by which it is attached to the cabin are 0·9-inch in diameter, and pass over sheaves above the second platform arranged in such a manner as to form a 3-purchase tackle. The connection of these ropes to the cabin is also through the safety gear, and they descend at each side of the lift or cabin track parallel with the main ropes.

Method of Action.—For lifting, the water under pressure is admitted to the top end of the cylinder and drives the piston down, the exhaust orifice at the bottom being opened. The two ends of the cylinder are connected by a circulating pipe C of nine inches internal diameter, at the lower end of which is placed the valve or distributor D (*Figs.* 1 and 7). For lowering, the top and bottom of the cylinder are placed in communication through the pipe C, the water being simply transferred from the upper to the lower side of the piston and the admission port left open. When the lift is at rest both the admission and circulation are stopped, and the apparatus is *locked.*

Distributing Valve.—The distributing valve D is shown in detail in *Plate* XII., *Fig.* 1. It consists of a vertical cylindrical valve-chest of nine inches inside diameter, in which works a hollow cylindrical slide-valve S and a piston-valve P, both attached to the same spindle, the spindle-valve being packed with cup-leathers. The slide-valve S controls two pairs of facing ports which communicate respectively with the pressure supply and with the top of the cylinder through the circulating pipe C (*vide* also *Plate* XI., *Fig.* 1). Communication with the bottom of the main cylinder is established through a lower port controlled by the piston-valve P, and below the latter the valve-chest is open for the exhaust.

In order to keep the lift at rest, the upper and lower ports are simultaneously covered by the slide and piston-valves respectively. When the valve is raised, the lower port is uncovered below the piston-valve, and the water is allowed to escape from the lower end of the cylinder; at the same time, the slide-valve opens both sets of upper ports and admits pressure to the upper end of the cylinder by which the piston is moved downwards and the cage lifted.

By lowering the valve so that the piston-valve is beneath the lower port, while the upper ports are uncovered by the slide-valve S, the exhaust orifice is stopped and communication is established through the interior of the hollow slide-valve S and the pipe C, between the top and bottom of the cylinder, the pressure on either side of the main piston being approximately the same. Under these conditions the cage descends and raises the piston.

Auxiliary Piston and Valve.—As the force required to move the distributing-valve is about 8,800lbs., a piston M, 11 inches in diameter, actuated by hydraulic power, and secured to the continuation of the valve-spindle, is employed to work the valve.

The motion of this auxiliary motor is controlled by a small hand-worked piston-valve V, 1¾ inches in diameter, in precisely the same manner as the distribution-valve controls the main piston.

Stiffening Piston-rods.—The two long piston-rods of the main piston are prevented from sagging by being made to work through a dummy piston U (*Plate* XI, *Figs.* 1, 4 and 5) inside the cylinder, and a sliding-block B above, coupled together by a rod half as long as the main piston-rods, forming together what is termed a sliding "spider." This spider travels through half the length of the stroke up and down; it is pushed up by the main piston, and downwards by the pulley truck.

Automatic Stopping.—The mechanism is automatically stopped when the cabin arrives at either end of its journey by means of a projection E (*Plate* XI., *Fig.* 2) fixed to the main piston, which throttles the corresponding port as the piston approaches the end of its stroke. On the lower side this projection is hollow, and a small opening is provided to prevent shock and afford relief by allowing water to pass through the piston, a small valve being provided for the escape of the fluid on the upper side.

Cages or Cabins.—The cabins (*Plate* XI., *Fig.* 8), with two rooms one above the other, accommodate 50 persons. They are supported by wrought-iron frames on wheels, running on a track parallel with the inner side of one of the main piers of the tower. In consequence of this, the inclination of the floor of the cabin changes during its trip, and to meet this the floor of each room in the cabin was originally formed of pivotted steps, the inclination of which could be adjusted by the conductor during transit to the angle required by means of a lever.

In practice, however, it was found that the device did not answer, and instead of it, a plain fixed floor has been substituted, with cross strips for affording a foothold as the angle varies.

Weight of Cabin and Truck.—The cabin and its truck, with safety appliances and other gear, have altogether a weight of 23,900lbs., the component of which, parallel with the 54° 34′ inclination of the lift track, amounts to 19,510lbs.

Weight of Counter-weight.—The counter-weight is 55,000lbs., equivalent at the same inclination to 44,970lbs., and taking into account the purchase of the tackle (1 to 3), capable of balancing 14,900lbs., neglecting friction.

Weight of Pulley Truck, Piston, etc.—The weight of the pulley truck and piston and piston-rods, which also acts in favour of the power, amounts to 33,060lbs., or, reduced for a stroke equal to the lift, $\frac{34,060}{12} = 2,755$lbs.

Weight of Passengers.—The weight of 50 passengers is estimated at 7,700lbs.

From the preceding data the unbalanced load can be calculated, and amounts to

$$23,900 + 7,700 - 18,333 - 2,755 = 10,512\text{lbs.}$$

Net Work.—The work required to lift this through 380 feet is 3,994,560 foot-pounds.

Water used per Trip.—The quantity of water used per trip for each lift is stated to be 1,728 gallons, or 17,280lbs.

Head of Water.—The head of water for working the lift, reckoned from the level of the reservoirs at the second platform to the discharge from the cylinder, is 393½ feet.

Available Power.—Consequently the available power is 6,799,680 foot-pounds.

Efficiency.—$\frac{3,994,560}{6,799,680} = 60$ per cent. nearly.

Hydraulic Efficiency.—The *hydraulic* efficiency is, of course, higher than this, and would be found by substituting for the net load in the numerator the unbalanced load *plus* the frictional resistance.

Pumps.—The water supply for the two Otis lifts in the Eiffel Tower is furnished by two pumps, each delivering 11 gallons per second at their ordinary speed, but capable of supplying 18 gallons per second at a higher speed.

Safety Gear.—For all indirect-acting passenger lifts, in which ropes or chains are used for supporting the cage, some safety appliance is indispensable, and although for the Otis lift this gear is not hydraulic in character, its construction is so ingenious that it deserves a brief general description; I shall not, however, attempt to enter into details.

The four ropes from which the cage or cabin is suspended in an ordinary Otis lift are guided in pairs by an iron yoke round opposite sides of the cage, and the ends are attached by links to a cross-piece upon which the cage rests (*Plate* XII., *Fig.* 2). The yoke and cross-piece are connected by tie-bars, and together form a framework surrounding the cage. This framework is furnished with four gun-metal guide sleeves, which slide on planed uprights of timber G (*Plate* XII., *Fig.* 3). The suspending links T are not directly connected to the cross-piece, but at each end of the latter the two links are attached to opposite ends of a lever J, which has its centre of oscillation X in the cross-piece, and thus acts as a balance beam. Should from any cause whatever any one of the wire ropes be stretched to a greater extent than the rest, the balance beam rises at one side or the other and causes a catch-wedge Q to grip the upright guides.

This is effected by means of a gripping lever LL′, which, when the balance beam rises through the stretching of a rope, is struck either on the arm L or L′, as clearly shown, and comes into contact with the wedge, which it forces up and causes to be jammed between the sleeve and upright. The gripping lever is keyed to a spindle A, which passes through to the opposite side of the cage and there carries a second gripping lever, of exactly similar construction, so that the jamming process is repeated at the other side.

Governor.—In addition to the safety device just described, the further precaution is taken of providing a centrifugal governor, driven by a light endless wire rope attached by means of an arrangement of levers to the cage. When a certain speed is exceeded, the governor brings into action two clamps which grip the light rope and make it lag behind the cage and act on the gripping levers LL′ already referred to, thus causing the wedges to act.

Speed of Lifts.—The lifts work at a speed of 394 feet per minute, or 6·566 feet per second.

EFFICIENCY OF HYDRAULIC LIFTS.

According to Sir William Armstrong, the efficiency of lifts ranges from 95 per cent. for direct-acting lifts to 50 per cent. for those in which speed-multiplying tackle, in the form of sheaves or pulleys, is employed.

Mr. Ellington, in a paper read before the Institution of Mechanical Engineers in 1882, states the efficiency of some of his balanced lifts to be as high as 85 per cent. The efficiency in this case is calculated as

$$\frac{\text{Weight lifted} \times \text{height}}{\text{Pressure} \times \text{area of ram} \times \text{stroke}};$$

it represents the efficiency during ascent only, and involves the assumption that the unbalanced weight of the cage represents the friction of the machine during descent, and that this friction is the same as for ascent.

SPEED OF HYDRAULIC LIFTS.

For passenger lifts, the speed ranges from 0·316 to 6½ feet per second, the last-named speed being that of the Otis lift in the Eiffel Tower. For goods, the speed may be somewhat higher than this, but in any case due provision for automatic stopping must be made. The speed of 6½ feet per second above mentioned is unusually high.

The usual maximum speed of loaded direct-acting passenger lifts is about one foot per second, but with the cage empty a speed of over 10 feet per second has been safely attained, and for goods, six to seven feet would not be too high.

The ordinary Otis lift works at about five feet per second.

THE CHIGNECTO SHIP LIFT.

An imposing application of hydraulic power on a large scale is embodied in the Chignecto ship lift now being designed and constructed by Messrs. Easton & Anderson, under Sir J. Fowler and Sir B. Baker, who are the Consulting Engineers to the Chignecto Marine Transport Railway Co.

Purpose of Lift.—This lift is intended to raise vessels bodily from the sea level on to a ship railway, by means of which they are transported across the isthmus of Chignecto, connecting Nova Scotia with the mainland, and also to lower them again from the railway to the water. There are two lifts of similar construction, seventeen miles apart, one at either end of the railway, on opposite shores of the isthmus. Each ship is placed on a kind of travelling cradle, and carried on a double line of rails.

Height of Lift.—Weight and Size of Vessels Raised.—The maximum height of the lift is 40 feet, and the greatest weight of any vessel lifted 2,000 tons, while its greatest length is about 200 feet over all.

Time Required.—The time required for raising (or lowering) a ship is 20 minutes.

General Design.—In *Plate* XIII. is shown the general design of one of the lifts in elevation, plan, and cross-section. The difference in the position of the two halves of the cross-section represents the difference of level of the lifts on opposite shores of the isthmus.

Gridiron.—The cradle—omitted from the illustration—carrying the vessel is supported on a gridiron, composed of longitudinal and transverse girders, all built up of steel plates and angles, and braced at the ends and in the middle, as shown. The bracing in the middle has been omitted from the illustration, but is similar to that at the ends.

The gridiron is suspended from long links, attached at their lower ends to the transverse girders, and at their upper ends to rams, working in vertical hydraulic cylinders.

Hydraulic Cylinders and Rams.—The cylinders are fixed to the sides of the dock or basin, opening at one end to the sea, into which the vessel to be raised is floated. By admitting water under pressure to the cylinders, or opening the latter to exhaust, the rams are lifted or lowered, and with them the gridiron and cradle. Before a vessel is admitted to the basin the gridiron is lowered sufficiently to allow a vessel to float in over the cradle; then pressure is applied to the rams, and the whole is forced upwards until the cradle is at such a height that it can be transferred from the rails on the gridiron to the railway on land.

It will be seen that in general principle the arrangement is very simple; but in carrying it out in practice a vast number of details are involved, on the design of which the success of the mechanism is dependent.

Simultaneous Rise of Rams.—It is obvious that, in order to avoid racking strains in the gridiron (and, consequently, also in the cradle and ship which it carries), all the rams must rise simultaneously and at the same velocity. If left entirely to themselves after the admission of water to the cylinders, they could not be relied on to do this, as some might be forced upwards in advance of the others, owing to inequalities in the frictional resistance.

Automatic Regulation not Adopted.—The question was discussed by the designers as to whether some automatic system of regulation should be adopted by means of which the advance of the rams could be controlled, but it was finally decided that such regulation could be effected with sufficient accuracy by manual labour.

Method of Regulating Motion of Rams.—To this end, the hydraulic cylinders and rams, 20 in all, are divided into groups of five, two on one side of the gridiron and two on the other. Each of these groups is supplied by a separate main from the valve-house, and from the main, branch pipes lead to the individual cylinders. One ram of each group has attached to it one end of a rope, which is led over guide-pulleys to the valve-house, where at its other end it is connected with an indicator.

Indicators to Show Position of Rams.—This indicator shows the position of the ram in question, and that of the rest of the group is assumed to be for practical purposes the same. It is the duty of the attendant in the valve-house so to regulate the valves that the indicators representing each group register the same vertical position of all the rams.

Adjustment for Unequal Weight at Ends.—In case the vessel is so placed on the gridiron that one end of the latter is loaded considerably more than the other, this arrangement might not afford sufficient facility for equalizing the motion of both ends. To meet this case, separate small pressure pipes are led from the valve-house to the end rams—one pipe to the two land end rams, and one to the two sea end rams, each pipe being controlled by a separate valve in the valve-house. When required, the end rams can be disconnected from their respective groups, and can then be worked separately by their own valves, so as to give more or less lifting power to whichever end needs it.

Final Adjustment by "Chocks."—When the gridiron has been raised to a little above the proper level, the final adjustment in height is effected by chocks of cast-iron, which are pushed under the ends of the girders (small hydraulic cylinders and rams acting through rods being employed for the purpose), and the gridiron is then lowered so as to rest on them.

Pumps.—The pumps for supplying the water under pressure are double-plunger ram-pumps worked from the prolongation of the piston-rod behind the cylinders of compound horizontal engines.

There are four double-ram pumps to each lift. The rams have a diameter of five inches, with a stroke of three feet six inches, and run up to a speed of 36 revolutions per minute.

Mains.—The water is pumped through a 6-inch cast-iron main to the valve-house, whence it is distributed by smaller mains to the groups of rams as already stated.

Pressure.—The pressure of water employed is 810lbs. per square inch.

Accumulator.—An accumulator is used in connection with the pumps to maintain a small reserve of power, and to act as a regulator.

Load.—The dead load, consisting of the cradle, girders, and rams, is 1,200 tons, making the total gross weight to be lifted 3,200 tons.

Hauling Cradle on and off.—Wire ropes running over pulleys worked by small hydraulic motors are employed for hauling the cradle on and off the gridiron. The cradle is connected with the ropes, when required, by means of a gripping apparatus in a manner similar to that adopted on cable tramways.

Material of Rams and Cylinder Dimensions.—The rams and cylinders are of special cast-iron and steel mixture, having a tensile strength of about 16 tons per square inch. Each ram has a diameter of 25 inches; the internal diameter of each cylinder, except where it fits the ram, is 26 inches, and the thickness of the metal three inches.

Connection of Rams to Links.—The end of each ram is spherical, and supports a short cast-steel girder, to the ends of which the suspension links carrying the gridiron are attached. The spherical form of the ram-head, fitting in a corresponding socket of the girder, allows the latter to adjust itself so that the load is equally distributed between the two links.

Guides for Transverse Girders.—Steel joists (of I-section) guide the transverse girders at the ends; these guides are supported at the bottom, in recesses at the sides of the basin, by the foundation masonry, and are held at the top by steel frames, partly cast and partly wrought.

Attachment of Cylinders.—The cylinders are steadied near their upper ends to strong iron castings let into the masonry of the basin, and the lower ends rest on flanges on the foundations.

Emptying Cylinders.—As the climate of Nova Scotia is in winter very cold, an ingenious device has been adopted for emptying the hydraulic cylinders of water, to prevent its freezing in them. This consists in forcing out the water by means of compressed air, supplied by an air-pump driven from an auxiliary steam engine, which also works the air-pump of the condenser. The compressed air supplied by the pump is delivered in the first instance into a reservoir, from which it then passes into the cylinders.

Allowance for Friction.—A calculation of the total pressure on the rams—810lbs. per square inch on 20 rams, each 25 inches diameter—and a comparison of this with the gross load to be raised, shows that the makers have allowed about 10·9 per cent. for friction, which should be ample.

Quantity of Water Required.—The quantity of water required for a full lift, without making any allowance for leakage, amounts to 2,726 cubic feet, or, allowing 20 minutes for the lift, at the rate of 2·271 cubic feet per second, equivalent to a velocity of over 11½ feet per second in a 6-inch main.

Power Exerted.—The total effective power exerted during a lift is about 481·8 horse-power, while the work done (in 20 minutes) in exerting a pressure of 3,549 tons through a lift of 40 feet is 141,960 foot-tons, or 317,990,400 foot-pounds.

In determining the effective power of the engines required for driving the pumps to work this installation, of course the friction of the water in the mains has to be taken into account, as well as the losses in the pumps themselves.

The wire ropes for hauling the cradle on and off the gridiron are formed in two parts, which can be disconnected. One of these parts is always on land permanently attached to the hauling machinery, the other rests on the gridiron, and when the latter is lowered, has to be disconnected from the landward portion of the hauling rope, the ends of which are in the meantime secured to brackets in order to keep them in place ready for coupling up when the lift rises again to the railway level.

CLARK & STANDFIELD'S CANAL LIFTS.

A very interesting device, which appears likely in the future to supersede to a great extent locks on waterways, is Messrs. Clark & Standfield's lift for canal boats or barges.

The barge to be raised or lowered is floated into an iron trough, which takes the place of a lock, and is, like the latter, closed at each end by gates. The quantity of water in the trough is only sufficient just to float the barge.

The trough is supported on the top of a single hydraulic ram which has a length of stroke equal to the difference of level through which the lift takes place.

The upper reach of the canal is extended by means of an aqueduct or embankment over the end of the lower reach, and is closed by a sluice-gate. When the trough is in its topmost position it abuts against the end of the upper canal, and when its up-stream gate and that of upper canal are open, forms a continuation of the latter, in and out of which boats can freely pass.

When the trough is in its lowest position it is submerged to a certain depth in the lower canal reach, which is here widened to contain it. In this case, when its down-stream gate is opened, the trough forms a continuation of the lower canal, and boats can likewise go in and out.

The ram and trough are balanced either by an accumulator of equal weight and stroke, or, where the traffic is sufficient, by an exactly similar ram and trough. Under the latter circumstances the top aqueduct is divided for a short distance into two branches, each with its own gate. The two cylinders are in communication with each other, and a comparatively slight addition of weight to either is sufficient to lower it and raise the other. This additional weight is given by always stopping the rising trough, so that its water level is a few inches below that of the fluid in the upper canal, which, when the gates are open, causes water to flow from the canal into the upper trough, and thus make it heavier than the lower one. This excess of water in the upper trough is discharged when it descends to the lower canal by stopping the trough so that the level of the water it contains is slightly above that of the lower canal.

In this manner a great saving in water is effected, only a sufficient quantity being required to overcome the friction and balance that portion of the ascending ram which projects above its fellow at the end of the stroke. The consumption of water is what it would be with a lock in which the lift was only a few inches.

Great economy in time is also effected by the use of this arrangement, the actual lift occupying only four or five minutes.

The first of these lifts was erected at Anderton, in Cheshire, on the Weaver, for establishing communication between that river and the Trent and Mersey Canal.

"LA LOUVIÈRE" LIFT.

A good example of one of Clark & Standfield's lifts (of which a general perspective view is given in *Plate* XIV.) is that constructed on the Mons and Charleroi Canal, connecting Charleroi and the centre coal basins of Belgium with the north of France and the industrial district around Mons. This double lift is the first of four, having a rise and fall of 50 feet 6¼ inches.

Height of Lift.—Dimensions of Trough.—The locks used on other parts of the canal were designed to have a length between gates of 141 feet, a breadth of 19 feet, and a depth of water over the sill of 7 feet 10½ inches. The trough for the reception of the boats in the hydraulic lift was, therefore, made of the same dimensions, the mean depth of the water in it being also 7 feet 10½ inches.

Attachment of Troughs to Girders.—The troughs are carried by lattice girders, as shown in *Plate* XV., *Figs.* 1 and 2. The side plates are rivetted to the main girders, while the bottom plates are rivetted to light longitudinal girders supported by a set of transverse girders. The upper canal is carried out from an embankment by an aqueduct 58 feet 5 inches long, constructed similarly to the troughs, and closed by a sluice-gate. The lower canal for a short length at its end is also lined with iron and furnished with a gate.

Diameter of Ram.—Each trough is carried by a ram of 6 feet 6¾ inches external diameter, to which it is connected by four strong girders forming the head. The ram is centrally placed under the middle of the trough so that everything is perfectly balanced. The ends of the upper canal and trough abut against each other on one side at the top of the lift, and the ends of the lower canal and trough do the same at the other side at the bottom of the lift.

Joints.—To form a water-tight joint, the abutting ends of the canals and troughs are slightly bevelled, so that a wedging action takes place between them.

In the La Louvière lift, indiarubber facing is employed on the bevelled surfaces.

Guides.—At each side, in the middle of the trough, is a guide, in a slot in which slides a long cross-head attached to the trough.

Weight of Water for Working.—In this lift the two rams, with their troughs, balance each other in the manner already described. The excess weight of water required in either trough to work the lift is found to be about 60 tons, equivalent to 10 inches of water over the surface of the trough. Of this, 50 tons are necessary to balance the extra weight of the projecting rising ram, leaving 10 tons for overcoming friction.

Weight of Trough, etc.—Speed.—The weight of each trough with ram and water is about 1,100 tons. The time occupied in a single lift is 2½ minutes, which, with a stroke of 50 feet, is equivalent to a speed of 20 feet per minute.

Compensation for Leakage, etc.—To replace leakage or raise a ram, should it have been allowed to sink too low, a small accumulator, sufficient for a stroke of two feet of the rams, is provided, which is supplied with water by pumps driven by a turbine, worked with water from the upper canal.

Pressure.—The ordinary working pressure is 470lbs. per square inch.

Construction of Cylinders.—The cylinders for the rams are of cast-iron, four inches thick, turned on the outside, with weldless steel hoops, two inches thick, shrunk on like coils on a gun.

The initial tension in the hoops is so adjusted that they set up a compression in the cast-iron of about the same amount, so that when working under pressure there is scarcely any stress at all in the cast-iron part of the cylinder, which acts chiefly as a liner to keep the apparatus water-tight.

Cylinder Cast in Parts.—On account of its great weight, it was necessary to make the cast-iron cylinder in several parts, each about seven feet long, and as flanges could not be cast on, the coils at the end of each 7-foot length were rolled with flanges like an ordinary tire. By these flanges the separate parts were bolted together.

Discharge of Water.—To pass the water contained in one cylinder —about 50 tons, or 1,800 cubic feet—into the other in 2½ minutes requires an orifice of considerable size, and as a matter of fact, a communicating pipe 10 inches diameter is used between the cylinders. In order to avoid weakening the cylinder by making a hole of this size in the wall, which would also involve an interruption of the hoops. the plan was adopted of forming a number of small holes all round the cylinder communicating with an annular pipe, bolted between two coils by means of a flange round the inside; in this flange are the apertures connecting the pipe and cylinder.

To the annular pipe is connected the 10-inch pipe through which the water flows to or from the twin cylinder.

The construction is clearly shown in *Plate* XV., *Figs.* 3, 4 and 5, which represent respectively a sectional elevation of one cylinder. a cross-section through one of the annular pipes and the orifices, and an outside elevation of the top of the other cylinder.

Size of Barges.—The barges used are 128 feet long and of 400 tons burden.

Necessity for Economy.—The necessity for adopting some method more economical of water than ordinary locks was very urgent, owing to the absence of any available water supply in the district traversed by the canal.

HYDRAULIC TOOLS.

These include rivetting machines, hydraulic punches and shears, forging and flanging presses, bending machines, and slotting and planing machines.

Advantages.—Where it can be economically applied, the use of water for working tools has undeniable merits. The hydraulic mains can be laid underground, and thus overhead shafting is dispensed with, whilst the floor space of workshops can be more advantageously utilized, since the position of the machine-tools is not dependent on that of the lines of shafting. Besides this, the danger arising from the use of belts and pulleys is avoided, together with the wear and tear of shafting and bearings. These remarks apply equally to the hydraulic working of cranes, lifts, and travellers in workshops.

Hydraulic Power not yet Applied to Rotary Tools.—So far as I know, hydraulic power has not hitherto been adopted for working lathes or other rotary tools by independent reciprocating motors, and probably, used in such a manner, it would be rather wasteful.

HYDRAULIC RIVETTING.

Hydraulic machinery for rivetting is now almost an indispensable portion of the plant required in all important shipbuilding establishments and works engaged in the construction of boilers, bridges, roofs, and rivetted structures generally.

Introduced by Tweddell.—The system of hydraulic rivetting was practically introduced by Mr. Ralph Hart Tweddell, who designed the first plant for the purpose in 1865.

The thickness of plates rendered necessary by the increased steam pressures employed in marine boilers gave an impulse to the adoption of such plant, as without it a steam-tight joint was very difficult to obtain.

Essential Construction of Rivetter.—A hydraulic rivetter consists essentially of a ram, which works in a cylinder, and carries at one end a die for forming the rivet-head. The necessary power is furnished by water under high pressure acting on the ram within the cylinder, and supplied through pumps and an accumulator.

STATIONARY RIVETTING MACHINE.

The construction will be best understood by reference to *Plate* XVI., *Fig.* 1, which shows (in longitudinal section through the cylinders) a stationary rivetter for marine boiler work.

This machine is fitted with an appliance for bringing together the plates to be connected, and keeping them in close contact while the rivetting is performed.

Moving Cylinder and Cupping-die.—The cupping-die E, for forming the rivet-head, is carried by a prolongation of the moving cylinder F, which works over a stationary ram C, fixed to the main frame.

Plate-closing Ram.—Within the moving cylinder F is formed a cylindrical space or bore A, in which works the plate-closing ram G. This ram carries an annular head H, encircling the cupping-die E, and set a certain distance in advance of it, this distance being fixed by a stop in the cylinder A, with which the ram G comes in contact when pushed forward.

Auxiliary Piston.—To bring the plate-closing tool, and with it the cupping-die, up to the work, and withdraw them again from it, a small auxiliary piston and plunger, working in the cylinder K, are used, the plunger being connected to the ram G by the snug M.

The front side of this auxiliary piston, that next the work, which has the smaller area, is always open through the pipe P′ to the accumulator, the pressure from which, therefore, tends to push back the auxiliary piston, and with it the ram G and the cylinder F. Assuming both cylinder F and plunger G to be pushed as far back as they can go; when water from the accumulator is admitted by a special valve A, through the pipe P, to the back of the auxiliary piston, the latter is driven forward and takes with it the ram G. This result follows because the total pressure on the back exceeds that on the front of the auxiliary piston.

Power-saving Appliance.—The ram G advances alone until it comes in contact with the stop in the cylinder A, and draws in water under a low head, from a small tank, to fill the space behind it in A. Then, in consequence of the stop, both the ram G and moving cylinder F are pushed forward together, the space between the stationary ram C and the cylinder F being filled with low-pressure water from the tank passing in through a check-valve by way of the admission port *a*, which latter also communicates with the main valve D.

Method of Working.—When the plate-closing tool has thus been advanced into contact with the work, communication between the accumulator and the back of the auxiliary piston through the valve A is closed, and pressure is admitted behind the ram G, through the valve B and pipe L; this causes the plate-closing tool to press the plates firmly together and hold them. Communication is next established by means of the valve D, between the space behind the cylinder F and the accumulator, and the pressure from the latter forces forward this cylinder, with the cupping-die, to form the rivet-head. As the total pressure on the cylinder F is, on account of its larger area, more than double that on the ram G, part of the water under pressure in the cylinder A is forced back into the accumulator, and is afterwards again available for doing work.

Total Pressure.—In the 100-ton machine, the total pressure for plate-closing amounts to about 40 tons, the remaining 60 tons being applied to forming the rivet-head. For very heavy work, however, the whole force of 100 tons can be brought to bear on the rivet if necessary.

Avoidance of Waste.—By filling the spaces between the rams and cylinders with low-pressure water from a tank while the tools are merely being moved up to their work, the waste of power which would be incurred by drawing water for the purpose from the accumulator is avoided.

Formerly, such rivetters were provided with an arrangement of adjustable stops, by means of which the stroke of the machine could be regulated to suit the variations in the thickness of plates and length of rivets.

These stops were attached to a rod connected through a lever with the moving cylinder, and automatically closed or opened the valve by coming into contact with a lever by which the valve was actuated. This device is now found to be unnecessary.

Plate XVI., *Fig.* 2, shows a general outside view of a stationary rivetter similar to that just described.

Such machines are made with gaps of 12 feet, and are capable of exerting a total pressure of from 100 to 150 tons on the rivet.

PORTABLE RIVETTING MACHINES.

For many purposes a fixed rivetting machine is inapplicable, as it would be impossible to get the work into the position necessary for using the machine. In such cases, and they are very numerous, portable hydraulic rivetters are employed. These can be suspended from cranes and travellers, or in some cases supported by special carriages adapted to the purpose.

Direct-acting Rivetter.—There are many forms of portable rivetters; one of these, known as a direct-acting machine, is illustrated in *Plate* XVI., *Fig.* 3.

Suspension from Hydraulic Lifting Cylinder.—It is here shown supported from a hydraulic lifting cylinder A by links attached to the frame at D. It can be easily detached and connected at other points, E or F. The lifting cylinder, which acts like a hydraulic jigger, is suspended by its chain B. The chain may be hung from a crane. Rivetters of this type are made with gaps of from four to five feet, and can put a pressure of from 25 to 35 tons on a rivet.

Rivetter with Curved Cylinder.—In another pattern of portable rivetter (*Plate* XVI., *Fig.* 4), the dies are carried at the ends of levers, which at their opposite ends are furnished, one with the ram, the other with a cylinder in which the ram works.

The ram and the cylinder is each cast in one piece with its respective lever, and both are curved to a radius equal to the distance from the fulcrum.

This construction is adopted to insure greater stiffness by avoiding loose connections between the levers and the ram and cylinder.

In the case illustrated, the machine is attached to a hydraulic lifting cylinder B by a quadrant A. The lifting cylinder is free to revolve on the axis CD, passing through the end of the quadrant; this allows the position of the machine to be changed without moving the hanging gear. A small ram working in an auxiliary cylinder connected eccentrically to the main cylinder pushes back the piston after a rivet has been closed, and opens the jaws of the rivetter ready for another stroke.

Modification of Boilers to Suit Hydraulic Rivetting.—Since the introduction of hydraulic rivetting, the construction of marine boilers has been modified in such a manner as to make the joints accessible to rivetting machines. This is illustrated by *Plate* XVII., *Fig.* 1.

Valves for Rivetters.—The valves employed both for stationary and portable rivetters are of the construction I have previously described as the Tweddell valve, in dealing with high-pressure hydraulic valves (*vide Plate* I., *Fig.* 2).

Largest Work hitherto Executed.—The largest rivets for which hydraulic rivetters have so far been used in actual practice have a diameter of 1¾ inches, and are required for shipbuilding.

All shipbuilding yards, locomotive and railway carriage works, and marine boiler shops of any importance are now furnished with hydraulic rivetting appliances, and in specifications for locomotives and wrought-iron or steel railway-carriage frames it is usual to stipulate that all the rivetting, where possible, shall be performed by hydraulic power.

The work done by hydraulic rivetters is better and more uniform in quality than that resulting from hand-rivetting. The rivet-holes are filled by the rivets with greater certainty, and the plates and heads more effectually closed up. Besides this, it is possible by hydraulic power to do work which, without it, would not be attempted.

GENERAL ARRANGEMENT OF HYDRAULIC TOOLS.

A combination of stationary rivetter, accumulator, and hydraulic pumps is shown in *Plate* XVII., *Fig.* 2, where A represents the pumps, arranged vertically and driven by belting; B the accumulator, charged through the pipe C and valve D from the pumps; H the rivetting machine, to the cylinder E of which the water from the accumulator is admitted through the pipe F by the valve G.

Hydraulic Power in Workshops, etc.—In many workshops and shipbuilding yards, hydraulic power is employed in working tools, cranes and winches all over the establishment.

For instance, in a large and well-known steel works which I recently visited, engaged chiefly in the production of rolled tyres, cast-steel locomotive wheels, ingots and heavy steel forgings, hydraulic cranes and capstans are everywhere used, while a large hydraulic forging press is in course of erection. At an important locomotive building establishment in the same district, hydraulic power is similarly employed for cranes, capstans, turn-tables, lifts, and for pressing locomotive wheels on their axles. This last application of hydraulic power is universal in good locomotive works, and it is very usual for engineers to stipulate in their specifications that not less than a given total pressure shall be exerted for this purpose.

Thompson's Shipyard.—Plate XVIII, *Fig.* 1, represents the general arrangement of Messrs. Joseph L. Thompson & Son's shipyard at Monkwearmouth. A and A^2 are the pumps and pumping engines; B the accumulator from which the mains lead to C, where they branch off right and left to D and E. From the point F branch mains F^2 lead to the different shipbuilding berths. The exhaust pipes are in all cases laid alongside the pressure mains. When portable machines are used, flexible copper connecting pipes are required, which are shown at GG, etc., in connection with the machines they are to serve. The whole of the fixed piping is laid underground in trenches, and suitable stop-valves and hydrants HH, etc., are fitted where necessary. LLLL are cranes fitted with hydraulic lifts carrying rivetting machines of various types for doing different kinds of work; at K rivetting beams; at O floors and frames when made in separate sections. They are also placed at QQQ, at the heads of the building slips. At R, S and U are portable rivetters performing various operations; at WW are hydraulic winches placed on the ship's deck.

Plate XVIII., *Fig.* 2, illustrates a section of a vessel with various tools at work; at E and C portable rivetters, at H a rivetter making gunwales, at A a keel-rivetter, at G a stringer-rivetter, at M a winch, at K a portable punching and shearing machine.

FORGING AND FLANGING PRESSES.

For the very large forgings which are necessary for modern guns and marine engines and for the manipulation of armour-plates, steam hammers of adequate power would be enormously heavy, require very costly foundations, and occupy a large amount of space. The difficulties encountered in this direction by manufacturers have led in recent times to the adoption of the hydraulic forging press, in which the action of an enormous but steady pressure is substituted for the impact of a heavy weight.

As you are aware, it is impossible to arrive theoretically at the amount of the steadily applied force which is equivalent to that exerted by a falling weight of given mass, and, as far as I am aware, very little in the way of experimental data has been hitherto published on the subject.

Difference of Effects from Blow and Pressure.—In many respects the effects produced by a blow and by steady pressure are not directly comparable, especially with certain kinds of material, but, although it would scarcely have been safe to predict this on *a priori* grounds, it is quite certain that the work of the steam hammer can be satisfactorily performed by the forging press, while the latter occupies less space than the former, requires less expensive foundations, and produces no vibrations. As a matter of fact, the press is stated to make better forgings than the hammer, and to do its work, as far as results are concerned, with equal speed.

Difference in Handling Work under Hammers and Presses.—Experience has shown that under the forging press the work requires to be handled in a manner quite different from that adopted when a steam hammer is employed, the effect produced by the application of a steady pressure to a given part having been found to be very dissimilar to the result of a blow.

Flanging Presses.—Flanging presses are, strictly speaking, only a variety of forging presses adapted to the special work of flanging plates.

Presses for Light Work.—Hydraulic forging is not limited only to very heavy work, but, for producing some forms in wrought-iron and steel, the gradual action of the press is preferable to stamping by means of the hammer, and it is now very extensively used for light forgings. Among other applications may be mentioned the manufacture of shells and cartridge cases, the convex ends of which are dished out by hydraulic presses.

Steel Sleepers.—The steel transverse sleepers now so extensively used on the Indian State railways are formed at one operation out of a flat, soft, steel plate by a powerful hydraulic press, including the punched-up clips by which the rail-flanges are held. This affords a very good illustration of the accuracy attainable in this class of work, since the gauge is maintained by the clips, and only a very small error is admissible.

4000-TON FORGING PRESS.

A 4000-ton hydraulic forging press, made for Messrs. Chas. Cammell & Co., Limited, of Sheffield, by Messrs. Davy Bros., of Sheffield, is shown in *Plate* XIX., and will serve to explain the general construction of this class of machine. Two main pressure-rams, 36 inches in diameter, working in the vertical hydraulic cylinders CC, act through spherical-ended thrust-rods TT on the cross-head or tool-holder A. This cross-head is of inverted T form, guided at the ends of the arms by slide blocks, bored to fit four columns D, and at the upper end of the shank by a bored guide, bolted to the frame B, which carries the cylinders.

Dimensions.—The frame B is supported by the wrought-stee columns D, 20 inches in diameter, and by them secured to the base-plate E, which latter is formed of massive girders similar to those used for the frame B, but heavier.

The clear height between the frame and the bed-plate is 21 feet, the distance between the columns 15 feet in one direction and six feet in the other. To raise the tool clear of the work when a stroke of the pressure-rams has been completed, two lifting-rams F, nine inches diameter, are brought into action. These are attached by connecting rods G to the cross-head, which they lift. The full stroke of both pressure and lifting-rams is seven feet.

Ball-and-socket Joints for Guides.—The arms of the cross-head are connected to the guide-blocks by ball-and-socket joints, allowing a little play to accommodate the expansion that results from the heating of the cross-head when in proximity to the hot forging; the thrust-rods T, with spherical ends, are for the same reason interposed between the cross-head and the ends of the pressure-rams, and the cross-head is guided independently of the pressing-rams; this arrangement allows a forging to be placed considerably out of the central position between the rams without risk of grooving them.

The use of two pressing-rams instead of one reduces the width of the frame and allows the sling-chain carrying the forging to be brought closer to the anvil, when necessary, than would otherwise be possible.

Working Hydraulic Pressures.—The maximum hydraulic pressure employed is 4,500lbs. per square inch, but the water is supplied to the pumps from an independent source at a pressure of 60lbs. per square inch, and this low pressure is utilized to bring the rams down until the tool is in contact with its work.

Bringing Rams down to Work.—Assuming, for instance, that the cross-head is raised several inches above the forging; the outlet of the lifting-ram cylinders is first opened. The pressing-ram cylinders are thus placed in communication with the low-pressure water, and the action of this causes the cross-head to descend rapidly until the tool touches the forging. The operation can be performed in this way in much less time than if the water necessary for it were supplied by the pumps.

Low-pressure Valves.—The valves, which are of large size, communicating with the low-pressure service are now automatically closed and the pumps started.

Engines and Pumps.—Three pumps, worked from the crank-shaft of a pair of steam engines, supply water directly to the pressing-rams without the intervention of an accumulator. The pressure is automatically regulated by the resistance offered by the forging, and the consumption of steam by the engine also automatically adjusted to the work required.

The pumps have a plunger-diameter of six inches, with a stroke of 12 inches, while the cylinders of the steam engine are 34 inches in diameter.

The speed of the engines, controlled by an attendant, varies from 0 to 100 revolutions per minute.

Control of Valves.—The valves are controlled by a single hand-lever. U-leathers are employed throughout as packing.

The materials of which the press is formed are chiefly cast and forged-steel; the valves are of phosphor bronze.

The same water is used over and over again for working the press, and is contained in an air-vessel at the pressure—previously mentioned—of about 60lbs. per square inch.

The press in question is employed in the Steel Ordnance Department of Messrs. Cammell's works for large crank-shafts, guns, and other heavy work. As compared with a steam hammer, work can be got through more rapidly, and at the same time executed in a more thorough manner, by means of the forging press. As an instance of this, I am informed that the B-tube of a 68-ton gun, which would require about seven days to forge with the hammers, can be completed in two with the press. The original ingot for such a piece weighs about 50 tons.

FLANGING AND BENDING PRESSES.

Flanging and bending presses are necessarily very similar to forging presses, although, of course, the form of tool or die employed must be different.

Platen Flanging Press.—Plate XX., *Fig.* 1, makes the construction of one class of hydraulic flanging machine clear.

The lower frame or table B, carrying the hydraulic cylinder D, is connected by four columns C with the upper frame or table A.

The ram D^2 working in the cylinder D supports the moving head or table E, which is guided by the columns C. The table B also carries four auxiliary cylinders F, upon the rams G of which is fixed the platen H. To the table E is attached, by means of brackets, the mould or matrix J, formed to the shape of the outside of the plate to be flanged. Over this, in the proper position, the die K is fastened, also by brackets, to the upper table A.

The heated plate L is placed upon the platen H, and on water being admitted to the four cylinders F, their rams G move upwards and press the plate tightly against the die K. Water is then admitted to the main cylinder D and raises its ram and table E with the matrix J, forcing the latter against and over the plate L into the position shown by the dotted lines, thus flanging the plate.

Progressive Flanging Press.—In the flanging press just described a complete die and matrix of the full diameter of the plate to be flanged is required. For large work this method, of course, becomes very costly, and to avoid the expense entailed by it, the machine shown in *Plate* XX., *Figs.* 2 and 3, has been designed, in which the plates are *progressively* flanged.

This press has three hydraulic rams, which can be independently manipulated, two vertical rams A and C, and one horizontal D. The ram A is first made to descend, by admitting water under pressure to its cylinder, and holds the plate B fast upon the segmental die E, curved to the required radius. The ram C then comes down and bends the plate over, while finally the horizontal ram D is forced forward and squares the flange thus made. A short portion only of the circumference is treated at once in this manner. The plate turns upon a temporary centre, and successive portions are operated upon until the whole circumference is flanged.

For flanging and dishing the end of a boiler dome or similar work, the two vertical rams can be connected, as shown in *Fig.* 3, and used as one ram.

Hydraulic Punching and Shearing Machines.

Hydraulic machines for punching or shearing are necessarily constructed on much the same lines as stationary rivetting machines.

Plate XX., *Fig.* 4, is a vertical axial part section through, and *Fig.* 5 a side elevation of a combined punching and shearing machine, capable of punching 1¼-inch holes in plates 1¼-inch thick, or shearing plate of that thickness, or shearing angle-irons 6½-inch × 6½-inch × ⅝-inch. The piston B, working in the cylinder A, has an eccentric cylindrical stem C carrying the tool-holder D. The eccentric position of this stem brings the tool nearer to the front of the machine and prevents the piston from turning in the cylinder. The stroke of the piston can be regulated, to suit the thickness of the plate to be punched or sheared, by means of the adjusting nuts J on the rod H, which is connected, through the flexible joint I, lever G, rocking-shaft F, and lever E, to the stem C. The nuts J strike the valve-lever K.

The diameter of the piston is about 16 inches.

In section, the machine is shown with a punching tool attached.

LECTURE III.

HYDRAULIC MOTORS.

Generally speaking, the term "hydraulic motor" is applied to a machine used for the purpose of converting energy available in the form of water-power—whether as pressure or kinetic energy, or both—into such a form that it can be utilized for driving machinery of all kinds. In one sense, many of the machines with which I have been dealing might be considered as hydraulic motors, but they are never classed as such, being only applicable to special objects, and *not* for the general purpose of driving machinery.

Methods of Action.—For practical purposes, it may be said that there are three forms in which hydraulic power can be applied to the performance of work:—(1), as kinetic energy or through the velocity of the fluid; (2), by pressure; and (3), by weight.

Classification.—Each of these requires a different type of motor for its application; the *first*, the turbine; the *second*, the piston motor or pressure engine; the *third*, the water-wheel.

Time will only allow me to deal with the first of these types.

GENERAL PRINCIPLES APPLICABLE TO HYDRAULIC MOTORS.

With the elementary laws of hydraulics you are familiar, and it will only be necessary to briefly recapitulate those principles which are most essential to a comprehension of the action of water in hydraulic motors. The hydraulic phenomena chiefly associated with such motors are: the flow of water in pipes and channels; its outflow or inflow through orifices; and its action on curved surfaces.

Flow in Pipes, etc.—Hydrodynamic Equation.—To the flow of water in closed pipes and channels applies the law known as Bernouilli's, expressed by the well-known hydrodynamic equation, which is nothing more nor less than the application of the principle of the conservation of energy to water and the materials of the vessel enclosing it. It expresses the fact that, *provided there is true continuity of flow*, the sum of the potential and kinetic energy of the fluid flowing through any section of the pipe or vessel, at right angles to the directions of flow, is equal to that sum for any other section *plus* or *minus* any access or loss of energy from external sources.

The hydrodynamic equation takes the form

$$p+\frac{V^2}{2g}=p_1+\frac{V_1^2}{2g}+L,$$

where p and p_1 denote the hydrostatic pressures at two different sections, V and V_1 the corresponding velocities of flow, and L the access or loss of energy from external sources, as, for instance, by gravity in the form of increased head, or loss by friction, or by the performance of work on a motor.

The condition for continuity of flow is that an equal quantity of water per unit of time shall flow through every section of the closed pipe or channel, which must be quite full in every part.

If A and A_1 denote the sectional areas of the channel at two different points, measured normal to the directions of flow, and V and V_1 the corresponding velocities, while Q denotes the quantity of water flowing through per second, then the equation for continuity of flow is

$$AV=A_1V_1=Q.$$

Owing to this condition, when the dimensions of the pipe or channel, and the velocity through any section, are known, the velocity through any other section can be calculated, always on the assumption—not always fulfilled in practice—that the flow is uniformly distributed throughout each section.

From the hydrodynamic equation it is evident that as the velocity of flow *increases*, the pressure *decreases*, and *vice versa*. In a closed channel of varying section, therefore, the velocity must change to satisfy the requirements of continuity of flow, and with the velocity the pressure alters, so that a continual transformation is going on of kinetic into potential, and potential into kinetic, energy.

If A denote the energy available for any motor, W the effective work performed by it, and L the losses from all causes, then obviously

$$A=W+L.$$

The efficiency E is the ratio of the effective work to the available energy, so that

$$E=\frac{W}{A}=\frac{W}{W+L}.$$

Unless otherwise stated, under effective work I include the work done in overcoming the friction of the motor's bearings and the surface friction of the motor against the fluid.

TURBINES.

Definitions.—A turbine is a motor for utilizing the energy of water by causing it to flow over curved vanes, on which it exerts a reactionary effort constituting the motive force.

A turbine may also be defined as a water-wheel in which a motion of the water relatively to the wheel is essential to its action, in contradistinction to the old-fashioned water-wheel, in which such relative motion is not necessary.

Essential Construction.—A turbine consists essentially of a ring or pair of rings connected to a concentric shaft, and furnished with curved vanes arranged uniformly round the circumference.

On these vanes the reactionary force resulting from the flow of water over them is exerted, and causes the apparatus to revolve on the shaft. The rings are secured to the shaft by a boss and arms, like a wheel.

All modern turbines are also furnished with a stationary guide apparatus, containing ports or vanes which direct the water at the proper angle on to the wheel-vanes. The spaces between the wheel-vanes are often termed buckets.

Classification.—Turbines must be classified in two ways: (1). With regard to the *system*, or manner in which the water acts in them. (2). According to their mechanical construction.

In the first place, then, they may be divided into (*a*), *reaction*, and (*b*), *impulse* turbines, and each of these again sub-divided into *axial*, *radial* and *mixed-flow* turbines.

In *axial* or *parallel-flow* turbines, the water flows through in a direction generally parallel with the axis, or, more strictly speaking, parallel with a cylindrical surface concentric with the axis.

In *radial* turbines, the general direction of flow is radial with respect to the axis, and may be either inward or outward.

In *mixed-flow* turbines, the water enters radially and leaves axially. In practice I believe no impulse turbines are made with mixed flow.

ACTION OF WATER ON VANES.

Cause of Motive Force.—The motive force exerted by the water on the vanes of a turbine is due to its deflection by those vanes from its initial course. By the deflection, a retardation (or, relative to the wheel, an acceleration) of the velocity of the particles of water is produced, to cause which a certain force must be exerted by and against the vane.

To fix our ideas, let us suppose that a particle of water O (*Fig.* 12) is moving through the narrow curved channel *ab* with a certain velocity *v*, relative to the particular point of the channel over which it is passing, while simultaneously the channel moves with a uniform velocity *w* in the direction indicated by the arrow.

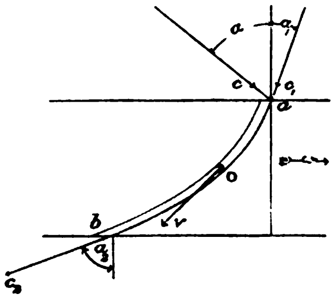


Fig. 12.

Illustrative Model.—This is illustrated in the model before you, in which a curved slot is formed in a thin board, which slides between straight guides attached to a second stationary board; in the slot moves a brush filled with ink.

The slot represents the curved channel, and the point of the brush the particle of water. As the brush moves down the slot—its motion being regulated by a lever with a counter-weight to prevent too high a velocity—the slide containing the slot is drawn forward by a cord passing over a pulley with a weight attached to the end.

It is obvious that the course of the particle relative to the moving slide—which represents a portion of the turbine—is always the same, being the central curve of the channel, but the path followed by the particle relatively to the stationary board, which, for the sake of distinction, may be called the *absolute* path, varies according to the velocity *v* of the particle along the channel, and the velocity of advance *w* of the channel. The brush traces the absolute path in ink on the fixed board.

Absolute Velocity.—The velocity *u*, relative to the stationary surface over which the particle passes while following the absolute path, may be termed the *absolute velocity.* It is easy to show that the absolute velocity is the resultant of the relative velocity *v*, and the velocity of advance *w* (also absolute) of the channel.

Let AB (*Fig.* 13) represent the absolute path of a particle O. At the point A its absolute velocity is *c*, at the point B this has changed to *u*, while its course has been deflected from the direction AC to the direction BD.

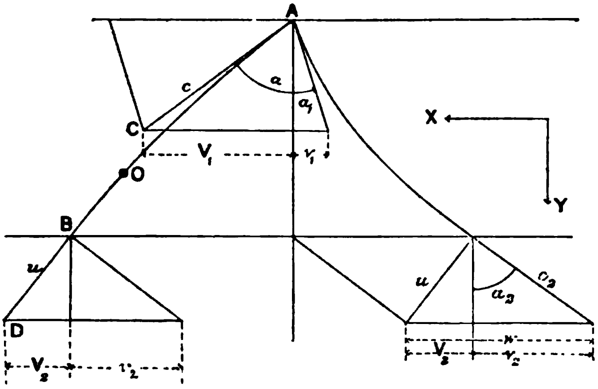


Fig. 13.

This deflection is due to the action of the curved channel, and therefore the force required to produce it is exerted against one side or the other of the channel (the sides being in practice the vane-surfaces of a turbine).

Forces Acting on Vane.—The resultant force at any moment can be resolved into two forces acting at right angles to each other; one X in the direction in which the channel is moving, the other Y in the direction in which no motion of the channel takes place.

The product of X and the velocity of advance *w* gives the work done per second by each particle of water. For a continuous series of particles all following the same course in the same manner, that is, a continuous stream of water, the total work done per second is the sum of all the products X*w*. It can be shown that when *w* is uniform, the total force exerted by the stream depends only on the difference in momentum of the water entering and leaving the channel measured in the direction in which the force is exerted. If *m* be the mass of water flowing through the channel per second, then the momentum at A on entering is the product of *m*, and the component of the absolute velocity *c* in the direction of the forces X; this component is V_1. Similarly, at B, the corresponding component of the velocity u is V_2.

Total Motive Force.—The total force exerted is, therefore—

$$X = mV_1 - mV_2 = m(V_1 - V_2),$$

and the work done per second—

$$W = m(V_1 - V_2)w.$$

When w is *not* constant, but increases or decreases uniformly from inflow to outflow, as in the case of a radial turbine, the work done depends on the total change of momentum and the maximum and minimum values of w. If these values are respectively w_1 and w_2, then

$$W = m(V_1 w_1 - V_2 w_2).$$

In any case, therefore, the work done is dependent only on the conditions existing at the points A and B of inflow and outflow, and *is not affected by the intermediate form of the channel* (*or vane-curve*). This, of course, is apart from any effect the form of the curve may have on the friction. For a radial turbine the components V_1 and V_2 are measured tangentially to the circumferences of the wheel.

The greatest deflection of the stream obviously occurs when the vane or channel is stationary, but in this case no work is done.

At a certain speed, on the other hand, the water leaves with the same momentum in the direction of rotation with which it entered. Then the resultant force exerted is *nil*, and under these circumstances also no work is done. This takes place, neglecting friction, when there is no resistance (or load).

Best Speed.—Between these two extremes there must be some speed at which the work done is a maximum, the so-called *best speed.* The best speed is theoretically about half that at which the wheel runs without a load.

The preceding rules and formula apply, strictly speaking, only to very shallow streams of water, but for practical purposes their use can be extended to channels or buckets of considerable depth, such as occur in actual turbines.

The centre line of the port or bucket is then taken as the relative path of the water.

Available Power.—The available power for any hydraulic motor utilizing a fall of water is, as you are aware, the product of the head or height of the fall (measured immediately above and below the motor) multiplied by the weight of water flowing through the motor per unit of time.

If h denote the head in feet, G the weight of water per second in pounds, then Gh is the available power in foot-pounds per second, or, expressed in horse-power—

$$A = \frac{Gh\,60}{33{,}000}.$$

If, as frequently happens, Q, the number of cubic feet of water per second, is given, then the available horse-power

$$A = Qh\,0{\cdot}1135.$$

In order that it may be able to escape at all, the water discharged from the turbine must have some velocity as it leaves the motor, and this velocity represents a certain amount of energy unutilized, and therefore, lost. If no friction or other losses occurred, the work done on the turbine would be simply the difference between the available energy and that unutilized. In fact, there are other numerous sources of loss.

Losses.—These are due chiefly to shock on entering the wheel-buckets, leakage, and friction in various parts of the apparatus.

Best Speed as Determined in Practice.—To obtain the maximum efficiency from any given turbine, it must be run at such a speed that the losses are reduced to a minimum, and in practice the best speed is determined from this consideration, which, in designing a new motor, governs the construction of the vanes.

The losses by fluid friction depend on the velocity of flow in the pipes and passages belonging to the motor, but as the velocity of flow generally varies only slightly or not at all with varying speeds of the wheel, the losses in question are also only slightly affected by the speed.

It is otherwise with the loss by shock and the loss by unutilized energy, and, therefore, turbines are so designed that when running at their normal number of revolutions both these losses are reduced to a minimum. Determined in this manner, the speed is a sufficiently close approximation to the best speed.

Inflow without Shock.—To insure inflow without shock, the water must enter the buckets of the wheel in a relative direction parallel to the surfaces of the vanes at the points of inflow.

This can be attained by making the velocity of rotation w, at the inflow, such that the absolute velocity of flow of the water entering is in magnitude and direction the resultant of the velocity of rotation and the *relative* velocity of inflow. This will be clear on reference to *Fig.* 14. OB represents the absolute velocity c, with which the water leaves the guide-passage (or port); OA is the velocity of rotation, and OD the relative velocity of inflow c_1, the *direction* of which is given by the wheel-vanes. OB must be the resultant of OD and OA. This is one of the conditions that should be fulfilled when the wheel runs at its best speed.

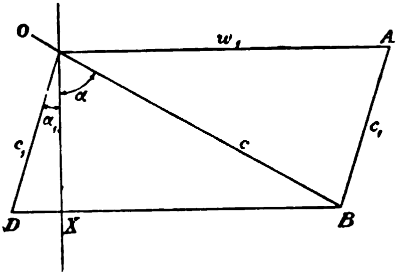


Fig. 14.

Velocity of Outflow.—As regards the loss by unutilized energy, the absolute velocity u, with which the water leaves the wheel-buckets, is the resultant of the *relative* velocity of outflow c_2 and the velocity of rotation w_2 at the point of outflow, as shown in *Fig.* 15, where OA $= u$, OB $= w_2$, and OC $= c_2$.

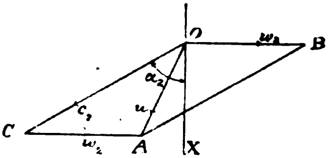


Fig. 15.

Vertical Outflow.—It is easy to see that with a given relative direction of outflow (OC), the absolute velocity u becomes a minimum when its direction is at right angles to OB the direction of rotation. To reduce the loss from unutilized energy to its lowest point, therefore, OA must be perpendicular to OB (*Fig.* 16). This is the second condition determining the best speed.

A turbine should have its vanes so constructed that the two conditions of inflow without shock and perpendicular outflow are simultaneously satisfied.

In a parallel-flow wheel, the outflow will be parallel with the axis, while in a radial wheel it will be radial.

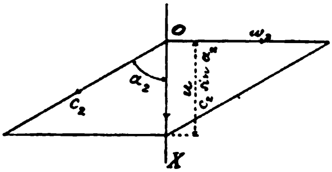


Fig. 16.

The mean velocities of rotation at inflow and outflow are the same in a parallel-flow turbine, while in a radial turbine they bear to each other the ratio of the radius at inflow to the radius at outflow.

REACTION TURBINES.

Essential Conditions.—In a so-called reaction turbine, it is essential that all the passages and every part of the apparatus should be *full* of water, so that there is everywhere continuity of flow, and the hydrodynamic equation applies. Without going into details, the effect of this is that the hydrostatic pressure of the water at the orifices of the guide-passages or ports, and at the inlets to the wheel-buckets, is in excess of atmospheric pressure, so that the velocity of flow from the guide-passages is very much less than would be the case if the water flowed out into the atmosphere. During the passage of the water through the buckets, this excess of pressure is expended in increasing the relative velocity of flow, and is thus finally utilized *indirectly* in driving the motor. It must, however, be distinctly understood that a reaction turbine is not driven by the hydrostatic pressure of the water in the same sense as a piston-motor.

Suction Tube.—A reaction turbine can be used in connection with a so-called draft or suction tube for the discharge of water from it, which enables the wheel theoretically to be placed at a maximum height of 34 feet (approximately) above the tail-water level. In practice, this height is always very much less.

Performance not Affected by Suction Tubes.—The performance of the turbine is not affected by its height in the draft-tube, within the limits admissible, or at any rate only to a very slight extent.

Continuity of Flow.—It must be understood that to the proper working of a reaction turbine, approximate continuity of flow is essential, and it follows that from the velocity of the water in any one part of the motor, that in any other part can be determined, with known dimensions.

Reaction Wheels Work Drowned.—To insure all parts being full and continuity of flow, reaction turbines should always work with the lower part of the wheel, or the end of the suction-tube—if one is employed—immersed in the tail-water; very often the wheel is entirely immersed.

As already mentioned, reaction turbines may be classified as axial, radial, and mixed flow.

Axial Turbines.—Jonval Turbines.—Axial, or parallel-flow turbines, are generally known as Jonval turbines, from the name of their inventor. One of these, with a double concentric set of guide-ports, is shown in *Plate* XXI., *Fig.* 1, in vertical section through the axis, while *Fig.* 2 shows a similar section of the buckets of an ordinary single Jonval turbine, and another section, developed in a plane, on a vertical cylindrical surface through the middle of the buckets.

Guide Apparatus.—Cover Plate.—The upper portion in all three figures represents the guide apparatus. In *Fig.* 1 this consists of three concentric cylindrical casings, between which the guide-vanes are secured, and a dished or conical cover, or plate, over the central space within the inner casing, having in the centre a water-tight bearing, or stuffing-box, through which the shaft passes. The cover may either be cast in one with the casing, or made separately and secured by flange and bolts.

Wheel.—The lower portion represents the wheel. This is formed by three concentric rings connected by the two sets of vanes which enclose the buckets, and attached by arms to a boss keyed to the vertical shaft. The guide-casing is supported on beams forming part of the floor of the reservoir or wheel-pit from which the motor draws its supply. The casing is let into a hole in the floor of the pit, and, of course, the joint between the casing and floor must be water-tight.

The water can be shut off from the guide-passages of the inner ring of the wheel by a series of covers actuated by rods from above. Sometimes every port can be closed separately, in other cases one cover serves for two or three ports.

Radial Outward-flow or Fourneyron Turbine.—A radial outward-flow wheel, which generally goes by the name of its inventor, Fourneyron, is represented in vertical section through the axis by *Plate* XXI., *Fig.* 3, while *Fig.* 4 gives two varieties of the form of vanes (A and B) used in such motors. The section through the vanes is at right angles to the axis of the wheel.

Treble Motor.—The turbine in question has a *treble* set of ports and buckets, practically equivalent to three separate motors, one above the other, with their wheels keyed to one shaft.

Regulation.—The ports can be shut off by a cylindrical sluice-gate, concentric with the shaft, and worked by a rod and lever from a hand-wheel above. With this method of regulation the guide-passages cannot be separately closed, but are all throttled to the same extent. The water enters the guide apparatus from below by a pipe, and flows outward through the guide-passages to the wheel. The wheel shown discharges above water, as is sometimes the case with Fourneyron turbines, but it is preferable with this, as with other types of reaction turbine, to work submerged or *drowned.* When the outflow takes place in the air, there is always a chance of the motor acting as an impulse wheel, and an element of uncertainty is introduced.

Buckets, etc.—*Divergent Passages.*—*Parallel Passages.*—With the form of vanes shown at A, the sides of the ports and buckets at the outflow are divergent, while with the construction illustrated at B, the passages between the vanes have parallel sides for a short distance at the outlet, with the object of insuring a perfectly definite direction of flow.

Radial Inward-flow or Francis Turbine.—*Plate* XXI., *Fig.* 5, will serve to explain the construction of a radial inward-flow turbine of the type known as a Francis or centre-vent wheel. The water enters the guide-passages from the wheel-pit on the outer circumference, and flows radially inwards. After leaving the wheel buckets, it is deflected downwards by the top of the wheel-casing, and issues into the tail-race by a short suction tube. The guide-vanes are pivotted at their outer ends, and the orifices of the guide-ports are regulated by swivelling the vanes about these pivots.

Plate XXI., *Fig.* 6, is a radial section through the ports and buckets.

Mixed-flow Turbine.—In a mixed-flow wheel, the form of the wheel-casing is much the same as in an inward-flow wheel, but the vanes are continued into that portion where the flow of water assumes an axial direction, so that the radial flow is changed into axial flow *while* the water is in the buckets, instead of *after* it has left them.

Fig. 17 illustrates the construction of one kind of mixed-flow wheel, without the guide apparatus, which latter is practically the same as for a radial inward-flow turbine.

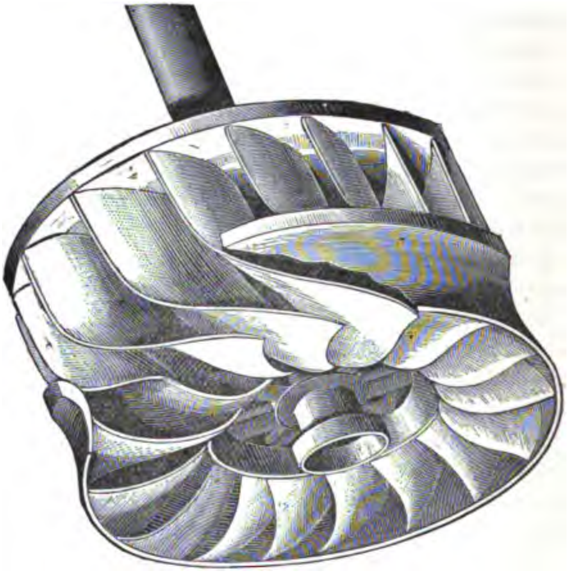


Fig. 17.

VELOCITY OF FLOW AND SPEED.

For the various kind of reaction wheels, as generally constructed, the relative values of the velocity of flow from the guide-ports, and of the speed of rotation at the inflow circumference for maximum efficiency, are much the same.

Velocity of Flow.—The velocity of flow from the guide-ports is, in designing a new turbine, one of the first quantities to be determined when those proportions and dimensions which can be arbitrarily fixed have been settled. On this velocity of flow depends the speed of the wheel and the area of the passages for a given power.

As a matter of experience, if h denotes the available head of water, and g the acceleration due to gravity, then the velocity of flow c from the guide-ports is :—

For Jonval turbines	$0{\cdot}67\sqrt{2gh}$.
„ Fourneyron turbines ...	$0{\cdot}75\sqrt{2gh}$.
„ Francis „ ...	$0{\cdot}725$ to $0{\cdot}64\sqrt{2gh}$.
„ Mixed-flow „ ...	$0{\cdot}64$ to $0{\cdot}612\sqrt{2gh}$.

These values apply to the effective area of the stream, and not to the measured area of the ports, which is greater. For instance, supposing the quantity of water used by a given motor to have been measured, the value of the velocity of flow, obtained by dividing this by the measured area of the ports, would be too low, as the flow is obstructed by the edges of the wheel-vanes passing in front of the ports; in radial-flow wheels allowance has also to be made for the contraction of the stream. The effective area may consequently be from 5 to 10 per cent. (or more) less than the measured area.

Speed.—The speed w of the circumference of the wheel over which the inflow takes place, measured in Jonval turbines to the middle of the buckets, has, in practice, the following values :—

For Jonval turbines, $w = 0{\cdot}64\sqrt{2gh}$.
„ Fourneyron „ „ $0{\cdot}625\sqrt{2gh}$
„ Francis „ „ $0{\cdot}64\sqrt{2gh}$.
„ Mixed-flow „ „ $0{\cdot}67$ to $0{\cdot}77\sqrt{2gh}$.

The preceding are normal values for turbines of the usual leading proportions, but it is quite possible to design reaction turbines so that the co-efficients are very different from those given. The values of these co-efficients depend chiefly on the ratio of the outflow area of guide-ports to that of wheel-buckets, and on the angles at which the vanes are inclined at inflow and outflow.

EFFICIENCY.

In practice, with good reaction turbines, an efficiency of 80 per cent. has often been obtained under favourable conditions, and is even guaranteed by some manufacturers. This is *exclusive* of the shaft friction and the surface friction of the wheel, which may amount to 3 or 4 per cent. and must be added in order to arrive at the hydraulic efficiency.

Carefully conducted experiments have resulted in efficiencies of 85 to 87 per cent.; but this is exceptional.

Full Gate.—The best results are almost invariably obtained when the motor is working with all the ports open, or at "full gate," as it is usually expressed.

Part Gate.—By partially closing the ports, the efficiency is reduced. The extent to which this reduction takes place depends very much on the system of regulation adopted, and it is generally more important to secure a good average efficiency at various gate openings than a very high efficiency at full gate.

An average efficiency of 70 to 72 per cent. when working with varying gate openings may be considered fairly good for a reaction turbine.

EFFECT OF SPEED ON PERFORMANCE.

Effect of Speed on Efficiency.—By running the wheel at either a higher or lower speed than the best, the efficiency of a turbine is reduced, but in the neighbourhood of the best speed the reduction is not very rapid.

Effect of Speed on Flow.—In radial inward-flow turbines the velocity of flow from the guide-ports decreases as the speed of the motor increases, while in radial outward-flow turbines the reverse is the case.

For Jonval turbines of the ordinary pattern the velocity of flow is practically constant at all speeds.

Curves Showing Effect of Varying Speed.—*Fig.* 18 gives, for an inward-flow wheel, the curves of efficiency and velocity of flow for varying speed. The abscissæ represent velocities of rotation in feet

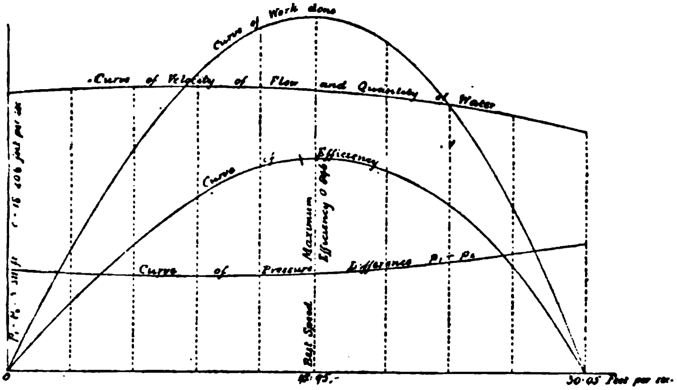


Fig. 18.

per second, the ordinates respectively efficiencies in per cent. and velocities of flow in feet per second.

FORM OF VANES.

In the design of the vanes, both of guide apparatus and wheel, the most important points are the angles formed by the vane surfaces with the direction of rotation and the ratio between the guide-port and bucket orifices. Provided the curvature is gradual, the intermediate form of the vanes between the points of inflow and outflow is of little importance.

Angles of Vanes.—The considerations which should govern the relative values of the vane angles have been already referred to; the angles should be such that the water enters approximately without shock and leaves vertically.

The angle of outflow from the guides and of relative outflow from the wheel can be independently chosen, within reasonable limits, as can also the ratio of the outflow areas. The relative angle of inflow is then fixed, and other values follow for a given quantity of water per second.

In Jonval turbines, where the width of the ports and buckets is made constant, a certain relationship exists between the angle of outflow from the guides and the relative angle of outflow from the buckets.

Angles: How Measured.—In a parallel-flow turbine the angles are measured in planes, tangential to a cylindrical surface, passing through the *mean* circumference of the wheel, and between lines parallel to the axis and tangents to the vane surfaces in the planes in question. In radial turbines each angle is measured in a radia plane, between a radial line and the tangent to the vane surface.*

The vane angle, at any point, is generally not identical with the angle of flow, which latter should be measured on the centre line of the stream; for this, of course, allowance must be made.

IMPULSE TURBINES.

Whereas in a reaction turbine the buckets must be full of water, generally under a pressure in excess of that of the atmosphere, in an impulse turbine the construction is such that the buckets are only partially occupied by the water passing through them, and the atmosphere has free access to the remaining space, so that the water in the bucket is, throughout, under atmospheric pressure.

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* Many writers make use of the angles complementary to those defined.

Partial Admission.—In some cases the water never acts on more than a part of the total number of vanes at one time; the turbine is then said to work with partial admission.

In impulse turbines the water in the wheel-buckets is allowed *free deviation*, that is, freedom in a direction at right angles to the direction of flow, to take its own course, so that the sectional area of the stream is not restricted by the vanes.

Buckets Ventilated.—To insure the existence of atmospheric pressure, the buckets of impulse turbines are generally ventilated, as shown in *Fig.* 19, by holes formed in the sides of the casing, admitting air from outside to each bucket. This device was invented by a French engineer named Girard, who was chiefly instrumental in introducing the impulse type of turbine.

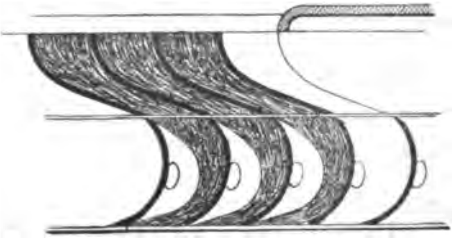

Fig. 19.

Impulse turbines are consequently very often described as "Girard" turbines.

The guide-passages are similar to those of reaction turbines, and should be full of water.

Wheel-buckets.—The wheel-buckets of impulse wheels are nearly always made wider at the outflow than at the inflow, with the object of allowing the water, during its progress through the buckets, to spread out so that each stream becomes shallower as it approaches the outflow. This permits the relative angle of outflow to be greater, without choking the orifice or increasing the pitch, and results in a higher efficiency than would otherwise be possible.

Axial Impulse Turbine.—*Fig.* 20 is a radial section through the buckets of an axial (or parallel-flow) impulse turbine, *Fig.* 19 a circumferential section through guide and wheel-vanes, and *Plate* XXII., *Fig.* 1, a general drawing of an axial impulse turbine with vertical shaft. The wheel is keyed to a hollow, cast-iron shaft, which runs on an overhead bearing, above water, of peculiar construction, very often used both for reaction and impulse turbines.

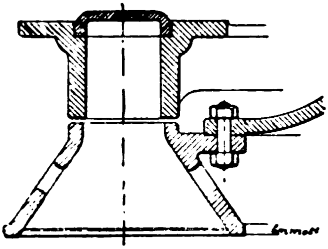

Fig. 20.

Overhead Bearing.—At the upper end of the cast-iron shaft is a sort of lantern, in which is secured a pivot, revolving on the top of a stationary wrought-iron spindle. The spindle is fixed in a socket secured to the bottom of the tail-race. The advantage of this type of bearing consists in its being easily accessible at all times.

Method of Regulation.—The guide-ports are opened or closed by vertical slides attached to rods actuated from above by a large cam. The cam, when caused to rotate, raises or lowers each of the rods in succession.

The water enters the turbine from a chamber or reservoir above the motor, into which the water flows under a sluice from the head-race.

Outward-flow Impulse Turbine.—An example of the radial outward-flow type of impulse turbine for small power and high speed is illustrated by *Plate* XXII., *Fig.* 2. It has a horizontal shaft.

The admission is partial, water entering the wheel through a few ports only, at the lowest possible position. The regulation takes place by means of a segmental slide, worked by a lever through a hand-wheel and screwed spindle.

Fig. 21 is a perspective view of a large turbine of similar type. Such motors are very well adapted for high falls and relatively small quantities of water, and are very extensively used in mountainous districts on the Continent.

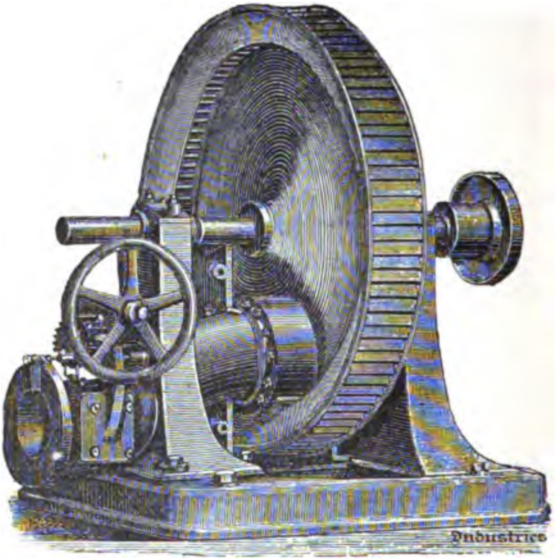

Fig. 21.

Tangent Wheel.—The radial inward-flow impulse turbine is generally known as the "tangent" wheel, on account of the water entering, in an approximately tangential direction, on the outer circumference. *Plate* XXII., *Fig.* 3, shows one of these motors, the invention of which is generally ascribed to a Swiss engineer named Zuppinger.

The water enters the wheel from a tank or reservoir through a pipe, at the mouth of which are three guide-passages which can be closed by a slide, worked by means of a rack and pinion. The axis is vertical.

Tangent wheels are now seldom constructed.

Ports and Buckets of Radial Impulse Turbines.—The general form of vanes and cross-section of guide-ports and wheel-buckets for a radial outward-flow "Girard" or impulse turbine can be clearly seen from *Fig.* 22, which relates to a motor of 580 horse-power.

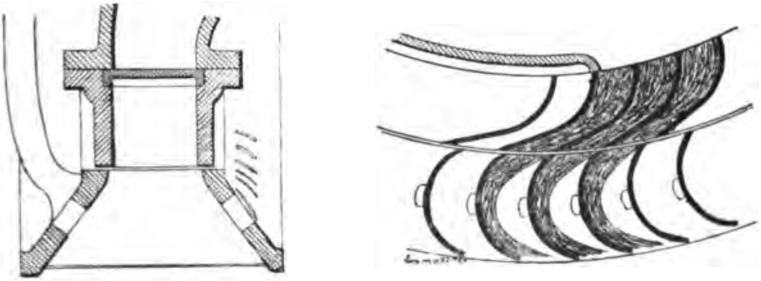

Fig. 22.

Pelton Wheel.—A type of impulse wheel, which is much used in the mining districts of America, is the "Pelton" wheel, or "Hurdy-gurdy" (*Fig.* 23). The shaft is horizontal. The buckets are double so attached to the outer circumference of the wheel that the water is deflected axially and escapes on either side. The water enters the buckets from a single nozzle near the lowest point of the wheel.

The construction is extremely simple, and efficiencies as high as 80 per cent. are stated to have been obtained with it.

Experiments with 700*lbs. Pressure.*—Experiments have been carried out by the Chester Hydraulic Engineering Company with a small motor of this type, driven by water from the London Hydraulic Power Company's mains, supplied under a pressure of about 740lbs. per square inch. An efficiency of 70 per cent. was obtained. The wheel has an outside diameter of about 18 inches, and ran at 2,000 revolutions per minute. The pressure indicated by a gauge close to the motor was 660lbs. per square inch.

The difference between this pressure and that of 740lbs. registered at the accumulator is due to loss in the pipes.

VELOCITY OF FLOW AND SPEED.

As there is atmospheric pressure at the guide-port outflow-orifices of impulse turbines, the velocity of flow is simply that due to the head, *less* the equivalent of losses arising from fluid friction and other causes in the reservoir, pipe, casing and passages, before the water enters the wheel, and is—unlike the velocity of flow for a reaction turbine—independent of what happens to the fluid after entering the wheel-buckets. For a certain fall, therefore, the velocity of flow is constant at all speeds of the wheel, and independent of its particular construction.

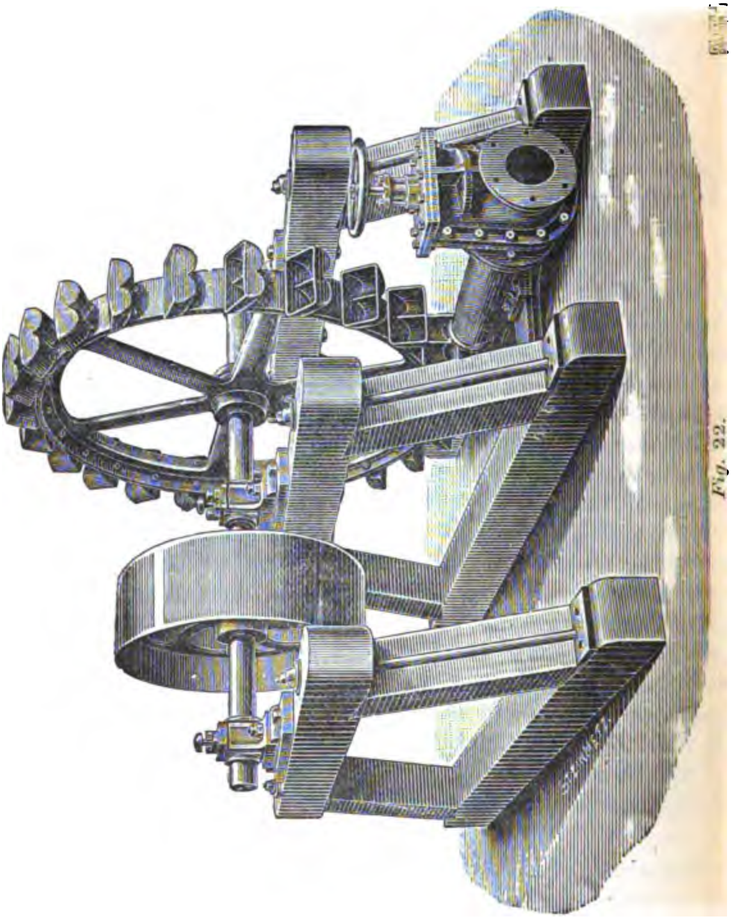

Fig. 22.

Discharge into Atmosphere.—To insure its proper action, an impulse wheel should discharge into the atmosphere and not under water; it must, therefore, be placed with the outlets of the wheel-buckets above the tail-water surface. This involves a certain loss of head h_2

for clearance, while, owing to the depth—in axial turbines—of the wheel, the orifices of the guide-ports are often still further above the tail-water by a height h_1. The effective head at the guide-orifices is consequently

$$H = h - (h_1 + h_2),$$

and the velocity of flow c ranges from

$$0{\cdot}9 \text{ to } 0{\cdot}95 \sqrt{2g\,H}.$$

The fall h_1, corresponding to the depth of the wheel, is utilized during the passage of the water through the buckets in accelerating the velocity of flow along the vanes, and thus increasing the work done, but the clearance h_2 is, of course, lost.

In practice, the best speed of impulse turbines, at the circumference where inflow takes place, ranges from

$$0{\cdot}45 \sqrt{2gh} \text{ to } 0{\cdot}50 \sqrt{2gh},$$

but is occasionally less.

The considerations which determine the best speed are the same as for reaction turbines, namely, inflow without shock, and radial or axial outflow.

EFFICIENCY.

Efficiency Lower than in Reaction Turbines.—The efficiency of impulse turbines is rather lower under the most favourable conditions than that of reaction turbines, but on the other hand, it is not diminished to the same extent by closing some of the ports; in fact, with well-designed impulse wheels the efficiency is very nearly as high at half as at full gate, and has in some cases reached 80 per cent., excluding the work spent on shaft friction.

This comparative constancy in the efficiency is the strongest point in favour of impulse turbines.

At speeds above and below the best, the efficiency decreases as in the case of reaction wheels.

FORM OF VANES FOR IMPULSE TURBINES.

The form of the vanes for impulse turbines is settled on the same principles which apply to reaction turbines, but, chiefly owing to the different proportions between the relative velocities at inflow and outflow, another form of curve for the wheel-vanes is the result. This will be very evident on comparing the preceding illustrations for the two classes of motor. The chief difference is in the relative angle of inflow, which, of course, affects the general character of the vane-curve.

Clearance between Water and Back of Vanes.—It is important that there should always be a certain amount of clearance between the concave surface of the stream flowing through a wheel-bucket and the convex or back surface of the adjacent vane.

COMPARISON OF REACTION WITH IMPULSE TURBINES.

Both systems of turbine—reaction and impulse—have their advantages and drawbacks, and are suited as a rule to different circumstances.

Conditions Suited to Reaction Turbines.—A reaction turbine is suited for utilizing water-power where the quantity of water available does not vary much or is always in excess of the maximum requirements, while the demand for power is nearly constant. When the power required is sufficiently great, a Jonval turbine may be employed for varying heads by dividing it into several parts, as previously explained and illustrated, using that of largest diameter, having the greatest velocity, for the maximum head, and the part of smallest diameter for the minimum head.

For quantities of water small in relation to the head, reaction turbines are unsuited, as they become very diminutive in size, while the number of revolutions is correspondingly great.

There is no absolute limit to the falls and quantities of water for which reaction turbines may be employed. The majority in use work with moderate heads under 50 feet, but have also been made for falls very much greater than this; they are also applied successfully to the utilization of falls as low as 18 inches.

In Europe the Jonval type of reaction turbine is the favourite, in Canada and the United States preference is given to the mixed-flow wheel; there is practically not much difference in the efficiency of the two kinds.

Conditions Suited to Impulse Turbines.—Impulse turbines have a slightly lower maximum efficiency than reaction turbines, but have the advantage that they will work almost equally satisfactorily at full and at part gate. The diameter is, beyond certain limits, independent of the head and quantity of water for which the motor is designed, and this allows a wheel, under any given conditions, to be constructed for either a high or low speed (number of revolutions), merely by making the diameter less or greater.

Impulse turbines are now to be found of all kinds, with vertical, horizontal and inclined axes, utilizing falls ranging from 4 to 1,800 feet, and quantities of water from 140 to 0·04 cubic feet per second.

Guaranteed Efficiency.—It may be mentioned that for both systems of turbine, under average circumstances, good manufacturers will guarantee an efficiency measured on the brake of 75 per cent., and a few even 80 per cent.

APPLICATIONS OF TURBINES.

Turbines of one kind or another are applicable in almost all cases where water-power is available, but, of course, before deciding whether such a motor is to be used in preference to a steam or other heat engine, the question of cost must be carefully gone into. Much depends on the preparatory works required in the shape of weirs, dams, canals and embankments, and very often, especially in this country, the interest on the cost of such works more than counterbalances the advantages to be secured by subsequent economy in maintenance, including avoidance of coal bills.

Electric Transmission in Relation to Water-power.—The introduction of electricity for the transmission of power to a distance has very greatly extended the possibilities for the employment of water-power.

In many districts there is a vast amount of water-power naturally available, but it is in situations far from industrial centres and cheap means of transport, and on that account has hitherto been neglected.

Electricity now enables us to transmit power, generated where it can be most cheaply obtained, perhaps in the heart of a lonely mountain gorge, and send it for miles through an electric conductor, into the midst of a great manufacturing town.

Keswick Installation.—A waterfall near Keswick has recently been applied to working a turbine which drives a dynamo, from which the current is conveyed for a distance of about three-quarters of a mile to the town of Keswick for the purpose of electric lighting. The electrical efficiency of this installation, which was recently described in a paper read before the Institution of Civil Engineers by Messrs.

Fawcus & Cowan, is stated to be 75 per cent., a figure which, however, I think must be rather too high.

The available head is about 20 feet, the quantity of water used at full gate about 23 cubic feet per second, and the available horse-power about 52.

UTILIZATION OF THE FALLS OF NIAGARA.

A gigantic scheme is now, as you are probably aware, under consideration for utilizing a portion of the falls of Niagara for motive power, through the medium of turbines. The undertaking is in the hands of the Cataract Construction Company. The total quantity of water estimated to pass over the falls is 265,000 cubic feet per second. From this it is proposed to draw off 10,200 cubic feet per second, with an available head of 140 feet. Assuming an efficiency for the motors of 75 per cent., the effective power developed under these conditions would be no less than 120,000 horse-power; yet this would represent less than 0·04 per cent. of the total flow over Niagara.

Two methods of carrying out this scheme are being worked out.

One of these consists in supplying the manufacturers of the district with water, at the necessary level, by means of a great central canal, and allowing them to erect hydraulic motors, which discharge into a tunnel forming the common tail-race; the motors would be also supplied by the Company.

The alternative scheme contemplates the erection of a central station, at which electricity will be generated by dynamos driven from turbines and distributed to various industrial districts, including the town of Buffalo.

For both schemes the underground tail-race discharging below the falls is necessary, and has now actually been commenced.

The tunnel will have a section of 490 square feet, and a slope of 7 feet in 1,000.

The form is rectangular, with a semi-circular arched roof, the height being 29 feet 5 inches, the width 18 feet.

EXAMPLES OF TURBINE INSTALLATIONS.

The following particulars and illustrations of arrangements of turbines for different purposes will serve to give you an idea of the various conditions under which these motors are applicable.

TURBINES AT OLCHING (GERMANY) (*Plate* XXIII.).

Here two axial or parallel-flow turbines, one of the Jonval (reaction), and the other of the Girard (impulse) type, work in conjunction, both on vertical shafts, and driving through bevel gearing one horizontal shaft. During nine months of the year the quantity of water available is from 565 to 600 cubic feet per second; in dry weather the volume sinks as low as 350 cubic feet. The fall is about nine feet six inches. The installation is so designed that with the usual quantity of water both turbines work together at full gate. When the supply diminishes, the Girard wheel is regulated by closing some of the guide-ports.

The manufacturer guaranteed an efficiency, ascertained by brake tests, of 75 per cent. with both wheels working together at full gate; 74·6 per cent. with the Jonval wheel working at full power and the Girard wheel at two-thirds gate; 73·8 per cent. with the Jonval turbine at full and the Girard at one-third gate; and 75 per cent. with the Jonval turbine alone at full gate. These guarantees were satisfactorily fulfilled during trials carried out by an expert.

The Jonval turbine is constructed with a double set of ports and buckets, partly with the object of strengthening the vanes, and partly to allow the buckets to be independently proportioned in the two parts.

The leading dimensions are as follows :—

	Inner ring.		Outer ring.
Mean angle of outflow from guides...	71° 10′	...	71° 10′
„ „ „ buckets	73° 25′	...	73° 20′
„ „ inflow into „	9° 25′	...	9° 25′
Clear width of buckets (inches) ...	16·34	...	13·78
Mean diameter	9ft. 6·3ins.	...	12ft. 3·6ins.
Area of guide-ports (square feet) ...	11·524	...	13·853
Outflow area of buckets „	9·253	...	12·002
Number of guide-vanes	23	...	30
„ wheel „	29	...	38
Thickness of guide-vanes (inches) ...	0·59	...	0·59
„ wheel „ „ ...	0·51	...	0·51

For the Girard wheel the dimensions are the following :—

Angle of outflow from guides	66° 0′
„ inflow into buckets	54° 0′
Clear width of buckets at top	11·81ins.
„ „ „ bottom	29·14 „
Mean diameter	9ft. 6·38ins.

Both turbines have overhead bearings, easily accessible from the floor above the wheels.

For the trial, the data and results are appended:—

	Both turbines together.	Jonval turbine alone.
Number of revolutions per minute, Jonval wheel	27·69	28·87
" " " Girard "	26·10	
Horse-power measured on brake	481·70	365·62
Available head (from head to tail-water) (feet) ...	9·433	9·991
Quantity of water used per second (cub. feet) ...	583·57	404·12
Available horse-power	624·72	458·33
Efficiency (per cent.)	77·1	79·8

Owing to the temporary incompleteness of the arrangements, the Jonval wheel during the trials worked 3·86 inches clear above the tail-water level, and no allowance is made for this in estimating the efficiency. If the 3·86 inches were deducted from the available head, then the efficiency would be 83 per cent. Apart from the unnecessary loss of head, it was a disadvantage for a reaction turbine to work above water.

JONVAL TURBINES AT ZÜRICH WATERWORKS (*Plate* XXIV.).

Two Jonval turbines, each with a treble set of ports and buckets, are used for driving some of the pumps for supplying water to the town of Zürich. The general arrangement is clear from the illustrations.

The three sets of ports and buckets are adapted for working with different heads and supplies of water according to the level of the river Limmat, from which the power is obtained.

The maximum head and quantity of water per second for which each ring of the turbine is intended is as follows:—

Outer ring: head, 10ft. 6ins.; water, 106 cub. ft. per second.
Middle ring: " 7 " 9 " " 141 " "
Inner ring: " 4 " 9 " " 233 " "

The power developed in each case is about nine horse-power at a speed of 25 revolutions per minute.

AXIAL IMPULSE TURBINE AT MELS (SWITZERLAND) (*Plate* XXV.).

This is a turbine in which the water is supplied to the buckets through only a few ports, that is, with partial admission. It drives a cotton-spinning and weaving mill. The buckets, as in all modern impulse turbines, are ventilated.

Regulation is effected in the first place by a gun-metal slide working the back of the guide-ports, and the speed is controlled by a governor acting on a throttle-valve in the supply-pipe.

The diameter of the wheel is 6 feet 10·68 inches, the width of the buckets at the inlet 2·76 inches, and the speed 225 revolutions per minute.

TURBINES AT GOKAK (B. INDIA) (*Plates* XXVI. and XXVII.).

These motors were erected for the purpose of utilizing the fall of the Ghatpraba river, which falls over a cliff near Gokak, in the Mahratta country. The maximum fall (in time of flood) is 180 feet 5 inches. The water is conducted from the top of the cliffs to the turbines below by a supply-pipe 32 inches diameter, which, for a considerable part of the distance, descends vertically, and then at an acute angle.

There are three turbines of the radial outward-flow impulse class with horizontal axes. Each wheel has a diameter of 67 inches, and develops 250 horse-power at a speed of 155 revolutions per minute.

The power is employed in driving a cotton mill of 20,000 spindles, to which it is transmitted by wire ropes, one from each turbine, having a diameter of one inch. The shaft of each turbine carries a rope driving-pulley 11 feet 6 inches diameter, from which the rope passes over carrier-pulleys 3 feet 3⅜ inches diameter, supported on masonry piers on the top of the cliff. The ropes travel with a velocity of 5,600 feet a minute.

The distance along the centre of the rope from the middle turbine to the driving-shaft of the mill is about 657 feet.

Branch pipes, of 24 inches diameter, lead from the main to each turbine, and can be independently shut off.

This installation was constructed and erected by Messrs. Escher, Wyss & Co., of Zürich.

TURBINES AT TERNI (ITALY) (*Plate* XXVIII.).

This is one of a series of radial outward-flow impulse motors, ranging from 20 to 1,000 horse-power, employed for driving the machinery of large steel works at Terni, all with horizontal shafts. The head of water available for all these turbines is 595½ feet. The particular turbine illustrated develops 800 horse-power, with a water supply of 15·89 cubic feet per second, and runs at a speed of 200 revolutions per minute.

The leading dimensions are as follows :—

Angle of outflow from guide-ports	70°
„ inflow into wheel-buckets	54°
„ outflow from „ „	70°
Internal diameter of wheel	8ft. 2·4ins.
External „ „	9ft. 5ins.
Width of guide-passages	4·91ins.
„ buckets at outflow	16·14ins.
Number of guide-passages	2
„ wheel-buckets	100
Maximum proportion of circumference over ... which water is admitted	$\frac{1}{36}$
Diameter of supply-pipe	24ins.

As this turbine drives a rolling mill, it is necessary to stop and start frequently.

To suddenly arrest the flow of water under such a high pressure as that used would involve great risk of bursting the pipes, and to avoid this an auxiliary branch-pipe, furnished with a relief-cock, has been provided. As the main-valve is closed, this relief-cock is simultaneously opened by gearing connected with the spindle of the main-valve.

The following data relating to the turbines at Terni will serve to give you some idea of the manner in which the type of wheel in question can be varied in diameter, irrespective of the head and quantity of water used, to suit practical requirements.

Horse-power.	Head in feet	Water, cub. ft. per second.	Revolutions per minute.	Inner diameter.	
				Ft.	Ins.
1,000	595½	19·77	180 to 240	7	10½
800	„	15·89	200	8	2·4
500	„	9·89	240	6	5·9
350	„	7·06	200	7	10½
150	„	3·00	250	6	4·7
50	„	0·99	850	1	10¼
40	„	0·85	450	3	6·1
30	„	0·60	600	2	7·5
20	„	0·42	450	3	6·1

The turbines at Terni were made by Messrs. J. J. Rieter & Co., o Winterthur, Switzerland.

TANGENT WHEEL AT ST. GOTHARD TUNNEL (*Plate* XXIX.).

This represents the general arrangement of a so-called tangent-wheel (radial inward-flow impulse turbine) employed at Airolo for driving air-compressors during the construction of the St. Gothard Tunnel.

JONVAL TURBINE FOR LOW FALL (*Plate* XXX.).

The turbine shown is of the ordinary parallel-flow type, but designed to work with a head of only 20 inches, about the lowest which it is possible to effectively utilize by means of a turbine. The power developed is about 80 horse-power, with a water supply of 300 cubic feet per second. An overhead bearing is employed.

HERCULES TURBINE.

Fig. 24 illustrates the arrangement of an American mixed-flow turbine, which, when tried at the testing station at Holyoke (in Massachusetts), gave very excellent results.

It is divided by transverse partitions into several parts, which can be successively shut off by a cylindrical sluice-gate sliding between the guide-ports and buckets.

GENERAL REMARKS ON THE USE OF HYDRAULIC POWER.

As I previously stated, no form of energy, with the possible exception of electricity, is capable of being utilized with the same degree of efficiency as hydraulic power under favourable conditions.

Nearly all those classes of hydraulic machines, however, which are actuated by the hydrostatic pressure of water exerted on a ram or piston, labour under the disadvantage that when worked from an accumulator or natural source of water supply, under a fixed pressure, they can only be used efficiently at *full power*. This is due to the practical incompressibility of water, which does not allow of its being employed expansively like air. The consequence is that the same quantity of water is required to fill the cylinder, whether the machine works at full power or runs without a load, the water consumed in both cases having the same value in available energy. This drawback is mitigated in some hydraulic machines by using two or more different sizes of rams, of which the larger act as cylinders for the smaller in the manner previously explained in connection with certain

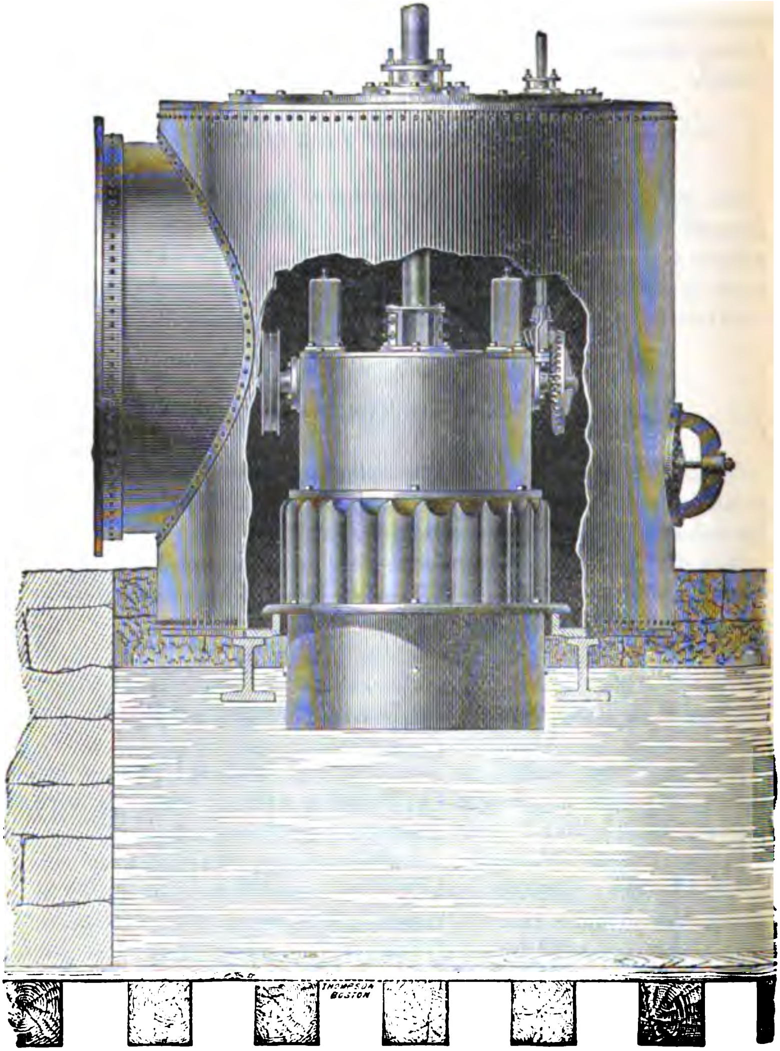

Fig. 24.

"jigger" cranes. Hastie's and Rigg's hydraulic pressure engines have been designed to overcome the defect in question by varying the stroke of the plungers.

In certain kinds of hydraulic presses, economy is obtained by the use of a ram having two diameters, and allowing the pressure to act in the first instance only on the area corresponding to the difference between the diameters, and afterwards on the full area, the space behind the smaller part being filled by low-pressure water to begin with. This is done in cases where the total pressure required during the first part of the stroke of the ram is comparatively small, and during the last part of the stroke much greater. The same end is also attained by employing several independent rams, to which pressure is applied in succession.

When the water is supplied direct to the working cylinder of a hydraulic machine, from pumps driven by a steam engine which can be economically governed by regulating the cut-off, the waste of power referred to does not occur. The pressure of the water then depends on the pressure of the steam in the engine cylinders, which is under control.

Turbines, if of suitable construction, do not lose very much in efficiency by working at part instead of at full gate.

There are many purposes to which hydraulic power is applicable besides those mentioned in the lectures which I am about to conclude, but it would have been quite impossible for me to deal, even in the most perfunctory manner, with any of these in addition to those I have described.

As it is, no one can be more sensible than myself of the somewhat sketchy manner in which I have been compelled by the exigencies of time to treat my subject, but I trust it may prove of some service in directing your attention to the important functions now performed in the mechanical world by hydraulic machinery.

It may possibly have appeared strange to some of you that I have not alluded to the application of hydraulic power to the working of guns; but this I have purposely avoided, as I believe you will all, as military engineers, have special facilities for acquiring a more thorough knowledge of this part of the subject than I possess.

In conclusion my thanks are due for the loan of valuable diagrams, models, samples, drawings and information to the following gentlemen and firms:—

Dr. W. Anderson; the American Elevator Company and their Manager, Mr. Gibson; Messrs. Clark, Standfield & Clark; the Institu-

tion of Civil Engineers and its able Secretary, Mr. James Forre
Messrs. Easton & Anderson; Mr. E. B. Ellington, of the Lond
Hydraulic Power Company; Mr. W. Günther, of Oldham; t
Institution of Mechanical Engineers and its able Secretary, Mr. Wal
Bache; Messrs. Nobes and Hunt (for leathers); Messrs. Jo
Russell and Company (for pipes and pipe connections); Messrs.
Simpson and Company; Mr. John Slater; Mr. Adolphus Steig
representative of Messrs. Escher, Wyss and Company, in Lond
Mr. Thornton, of the Hydraulic Engineering Company; and
R. H. Tweddell.

It now only remains for me to express my sincere thanks
Major-General Dawson-Scott, Major Kirke, Major Moore, and ot
officers, for their kind assistance and hospitality, and to you, gen
men, for your patient attention.

PLATE I.

Fig. 3.

Pressure Screw down Valve.
Scale 1/3.

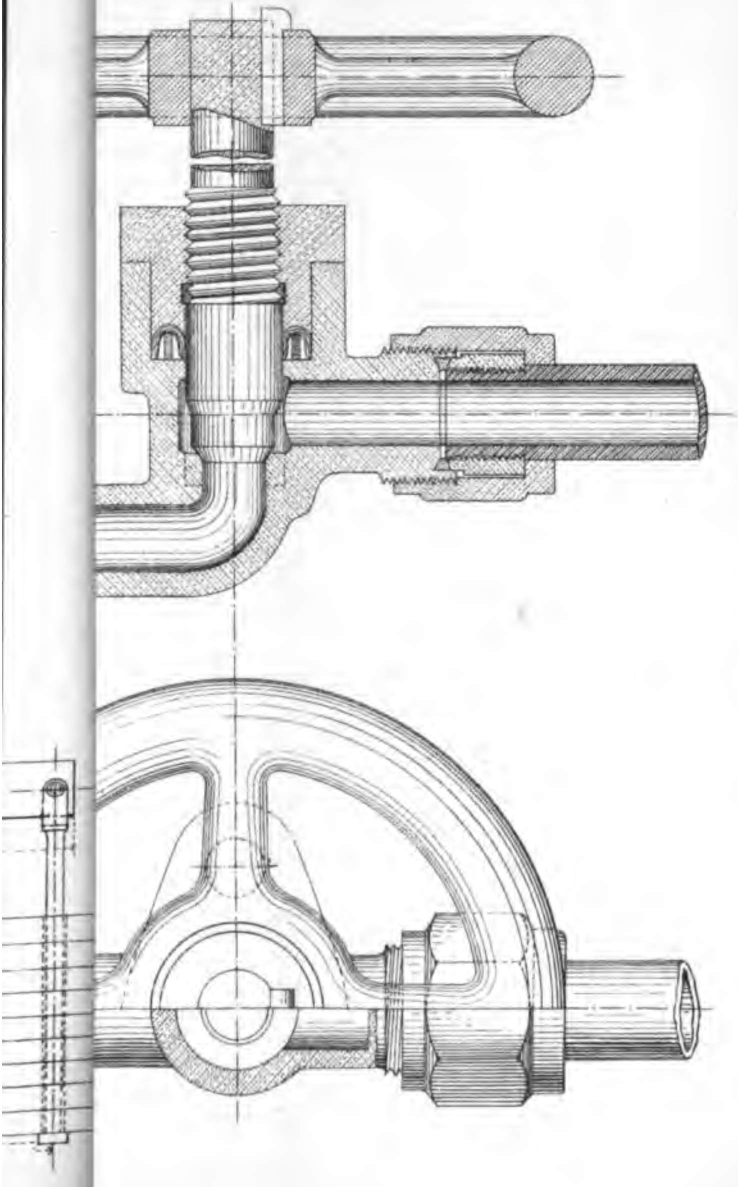

PLATE II.

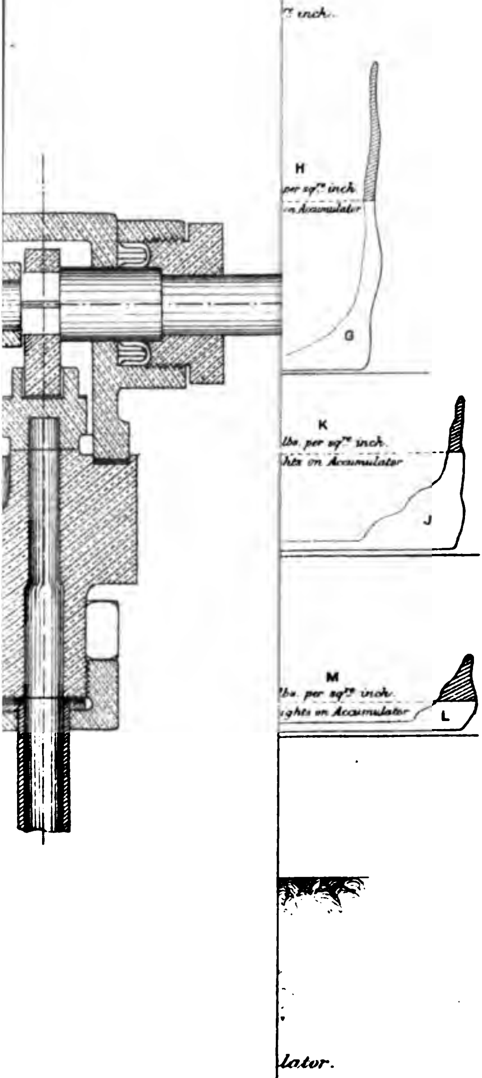

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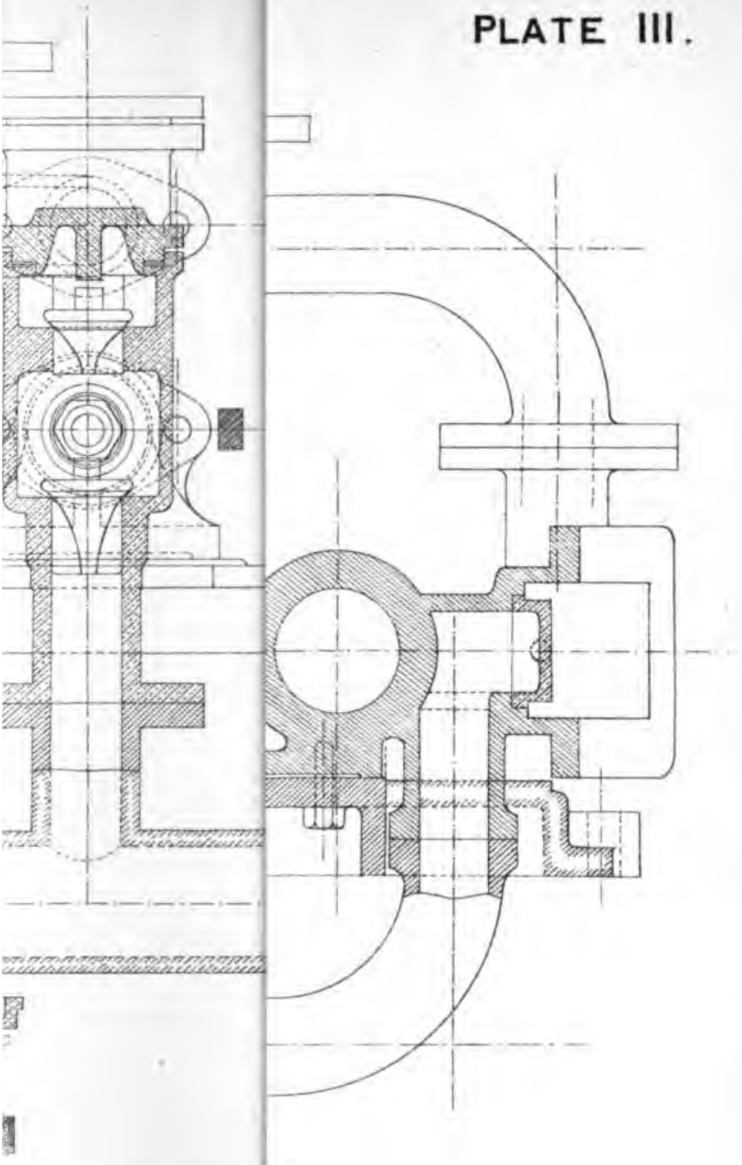
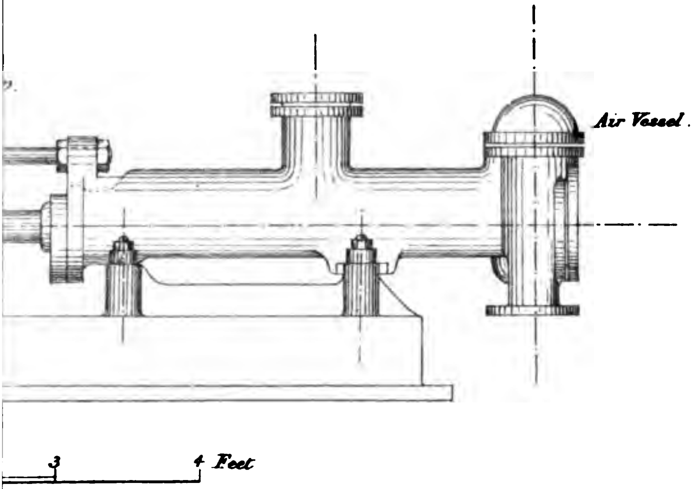
PLATE III.



on- to Slide Valve Con
ance valve. Scale, 1/8.

PLATE IV.



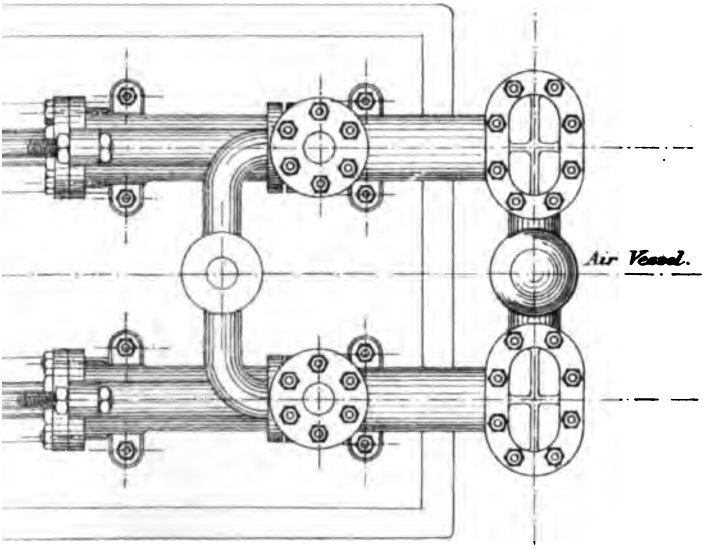


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PLATE V.

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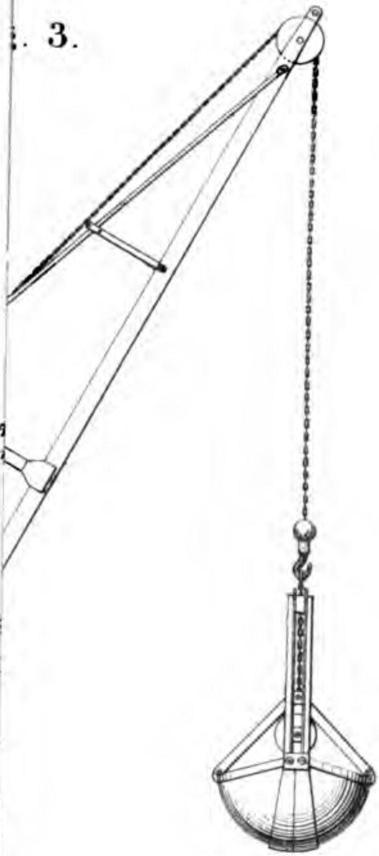
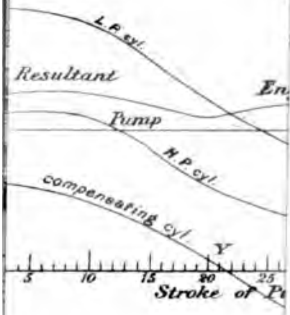


Fig. 2.

Mean of all diagrams f
A&B. taken at 12.30 p

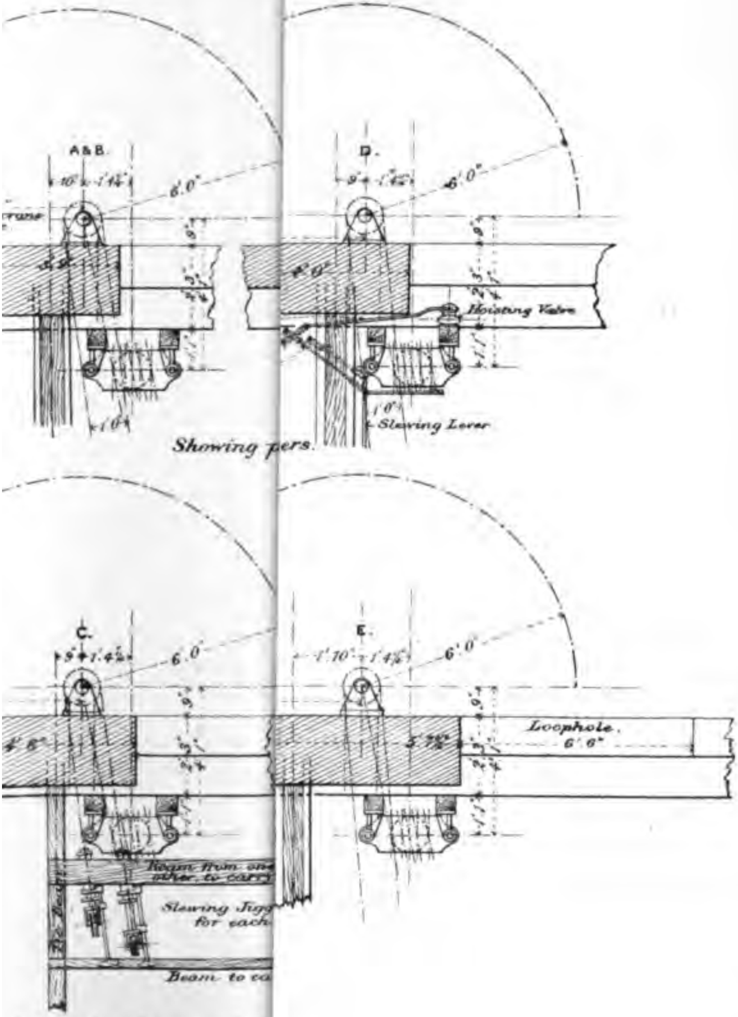


WER HYDRAULIC CR

Crosse & Blackwell.

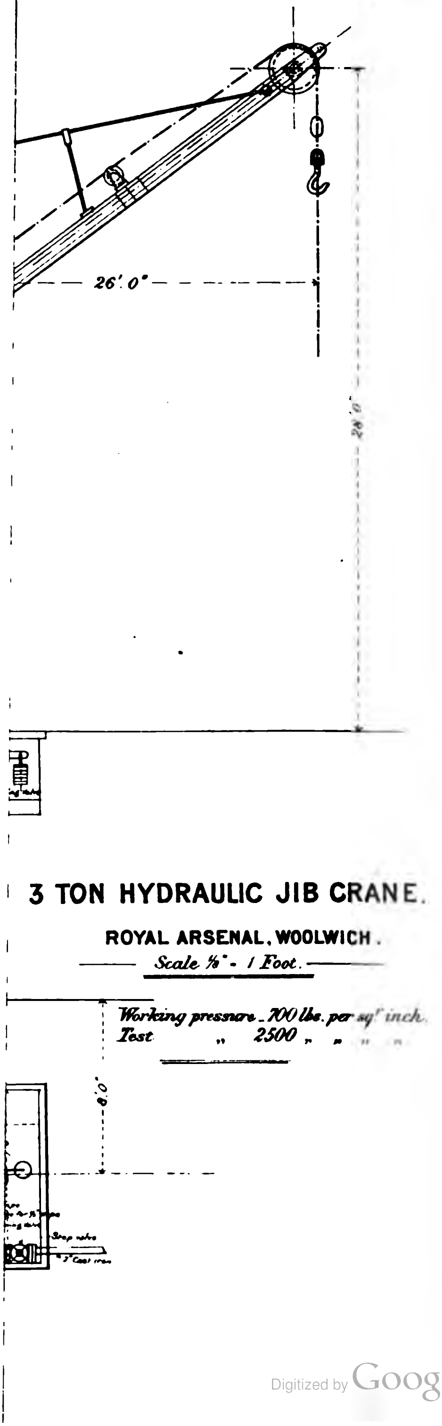
nent for fixing.

PLATE VI.



ber work to be provided

PLATE VII.



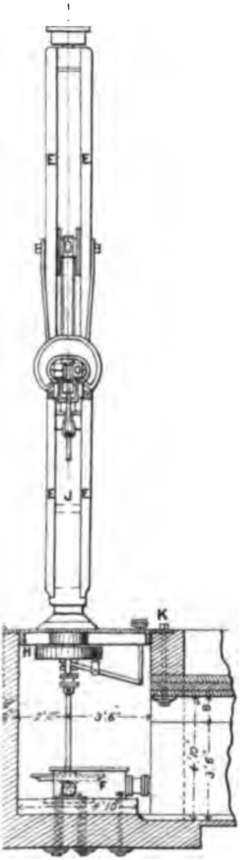
3 TON HYDRAULIC JIB CRANE.

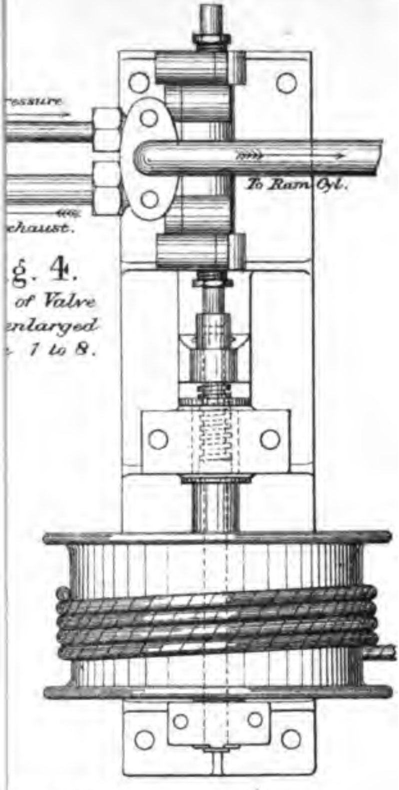
ROYAL ARSENAL, WOOLWICH.

Scale 3/8" = 1 Foot.

Working pressure . 700 lbs. per sq.^r inch.
Test " 2500 " " " "

PLATE VIII.

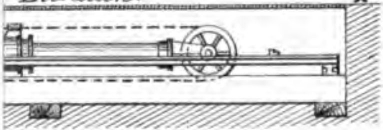




g. 4.
of Valve
enlarged
1 to 8.

gger Lift,
l cylinder.

Elevation.



Scale, 1 to 64.

Plan at XX.

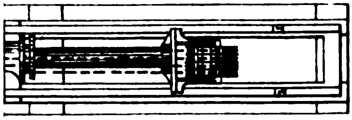
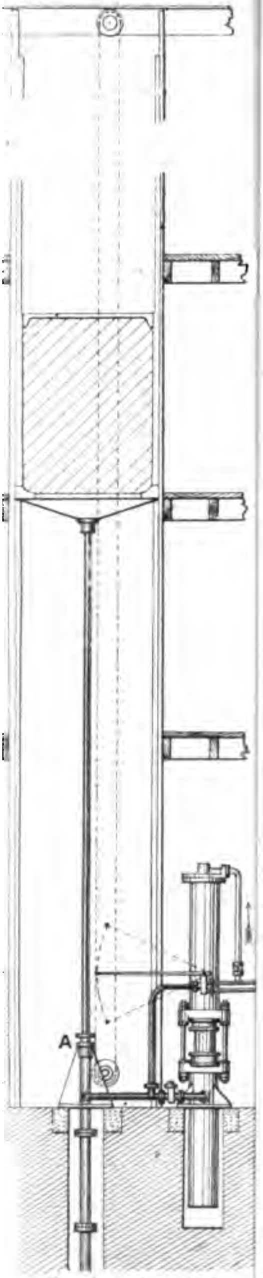


PLATE IX.



Scale, 1 to 96.

Fig. 6.

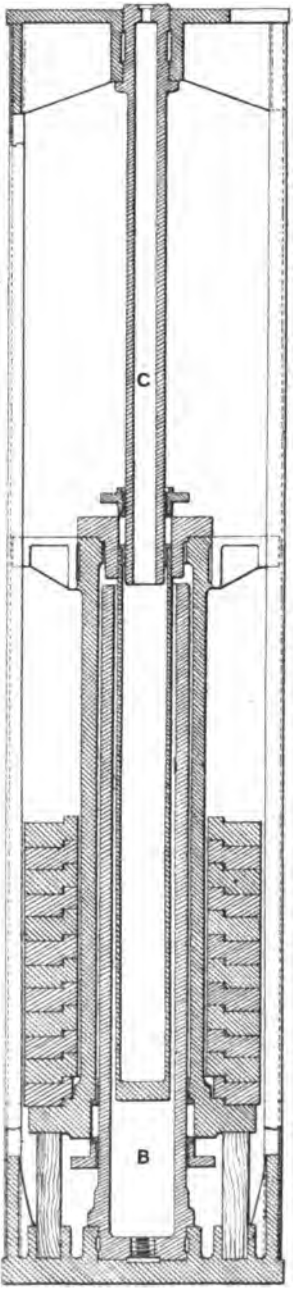


PLATE X.

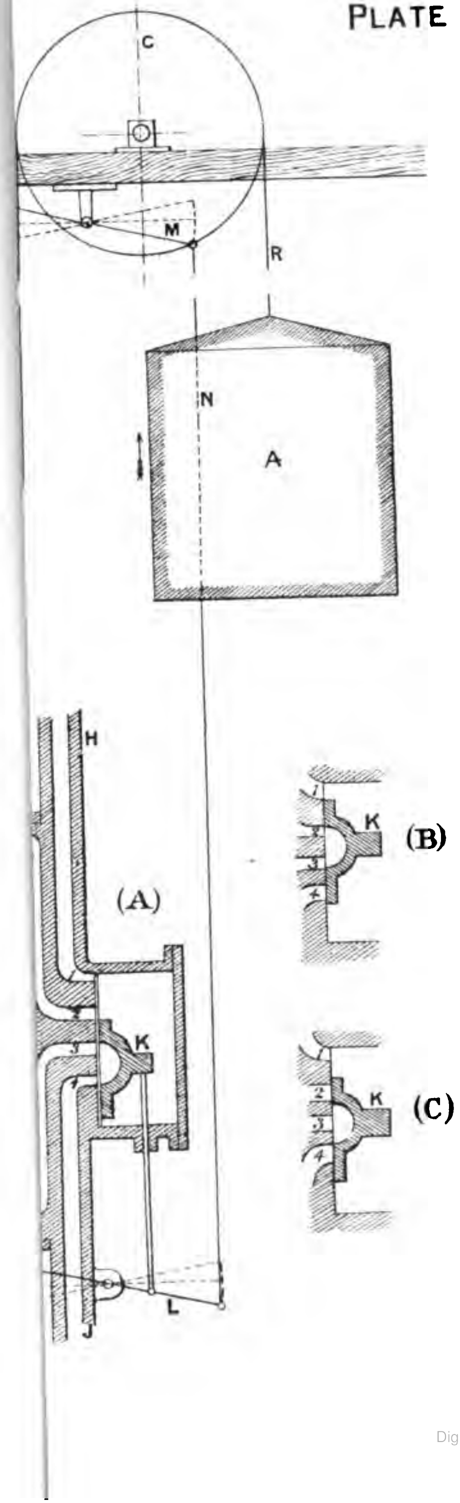


PLATE XI.

R LIFTS.

s.

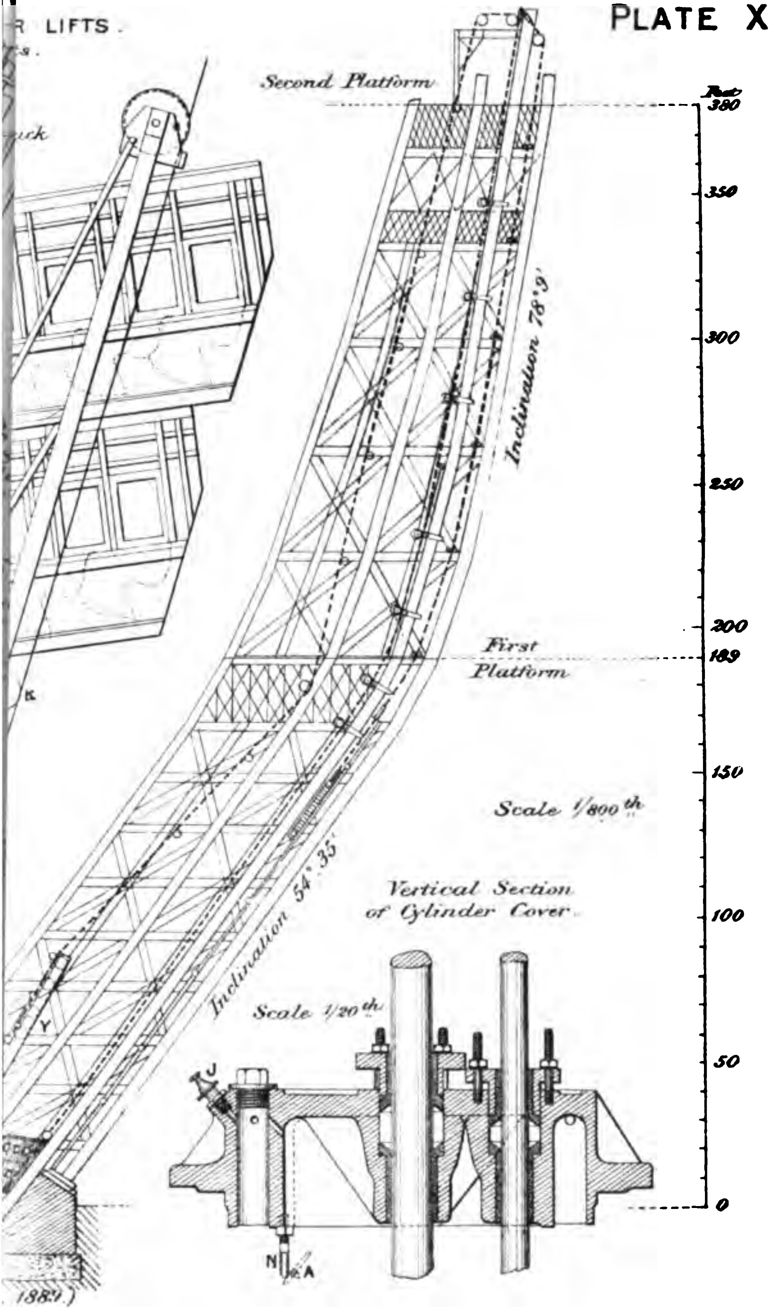
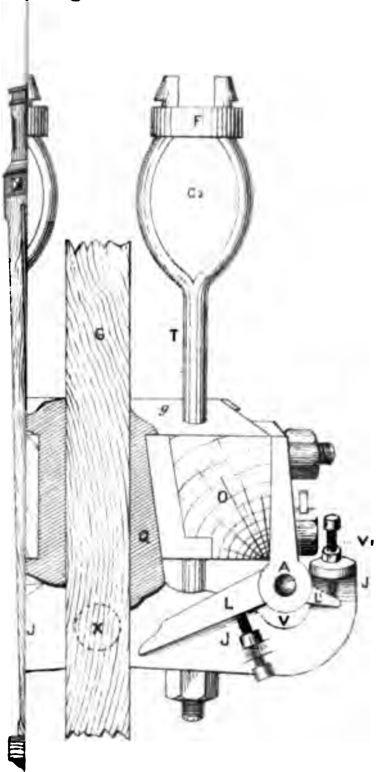


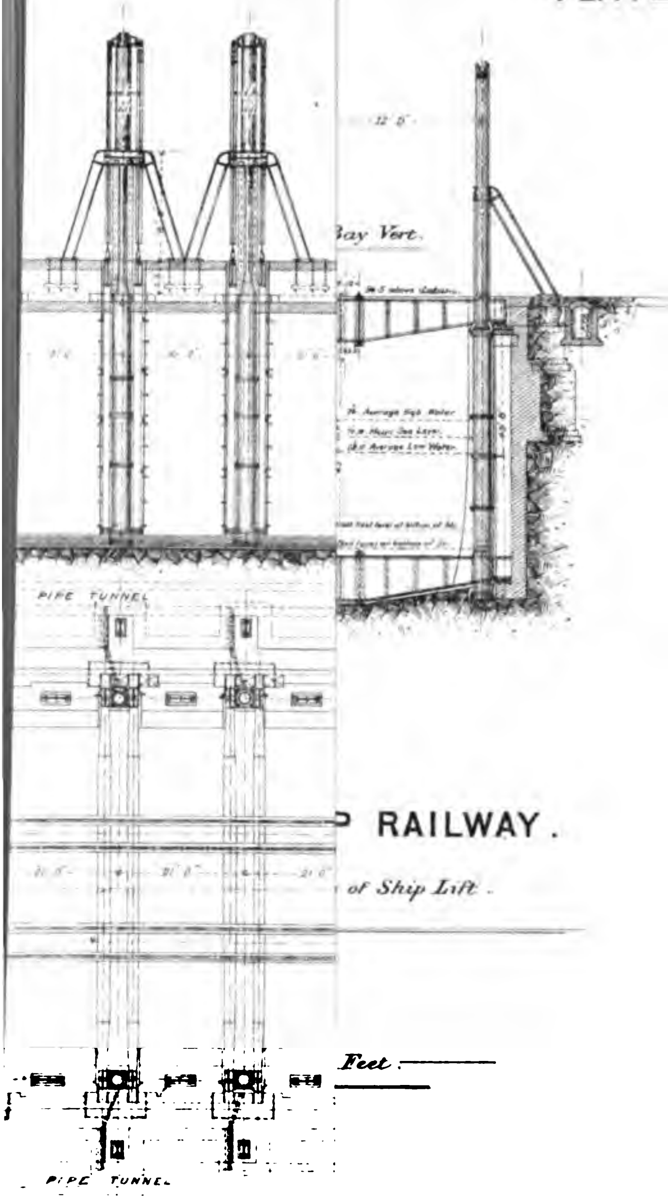
PLATE XII.

Fig. 3.



Feet.

PLATE XIII.



P RAILWAY.

of Ship Lift.

Feet.

PLATE XIV.

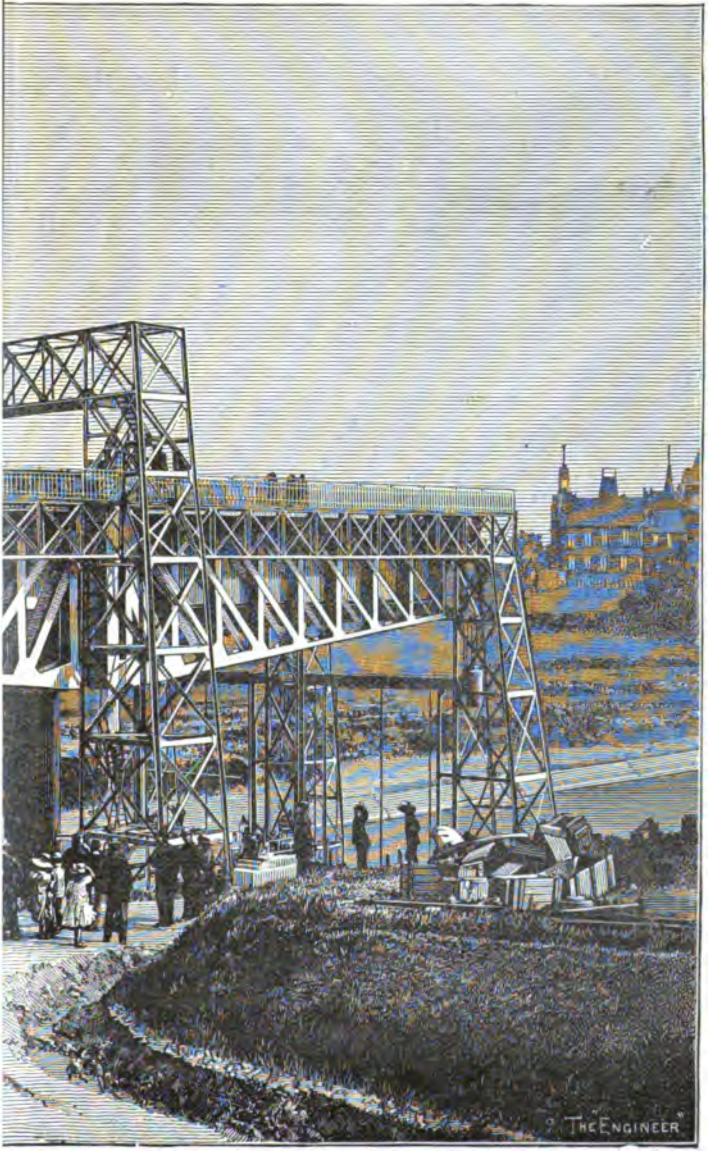


PLATE XV.

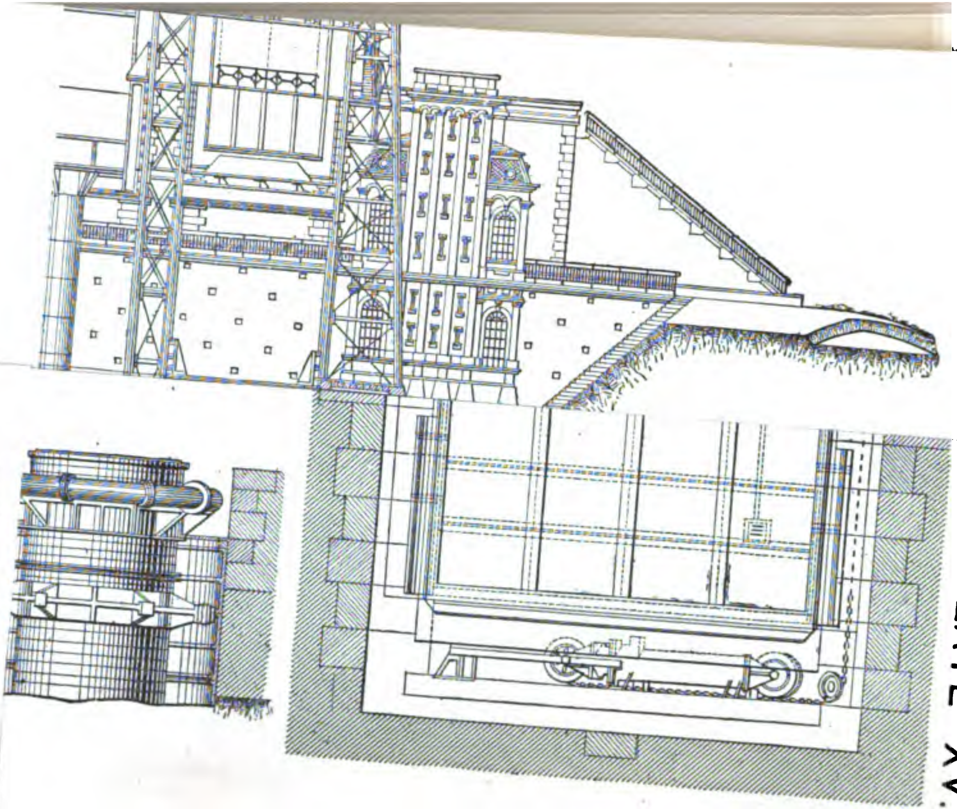


PLATE XVI.

Fig. 2.

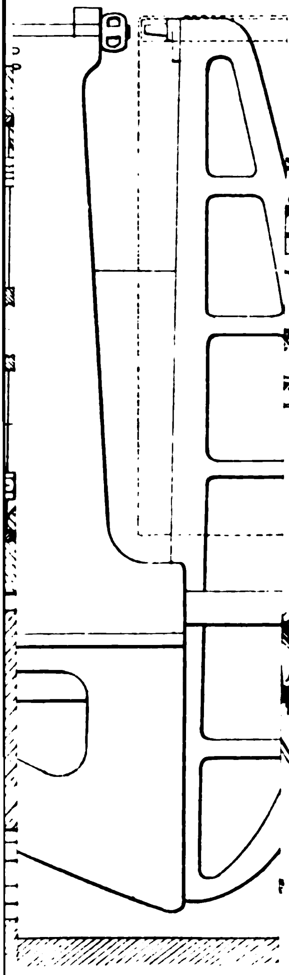
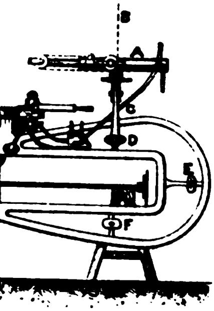
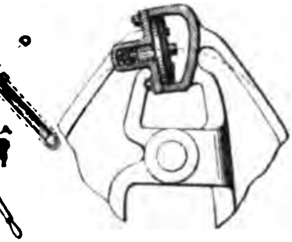


Fig. 3.



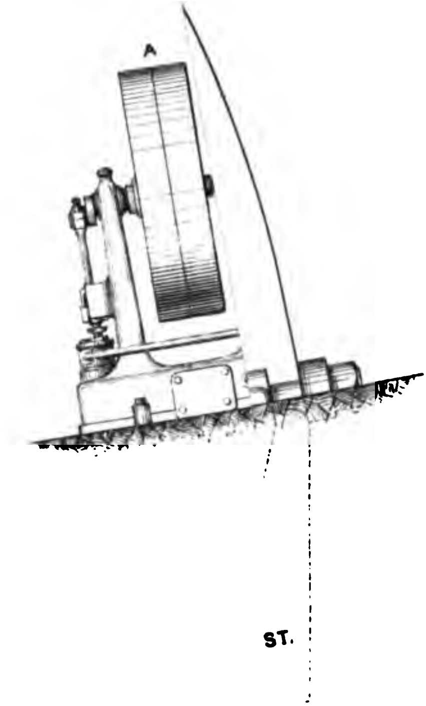
nch = 1 Foot.

Fig. 4.



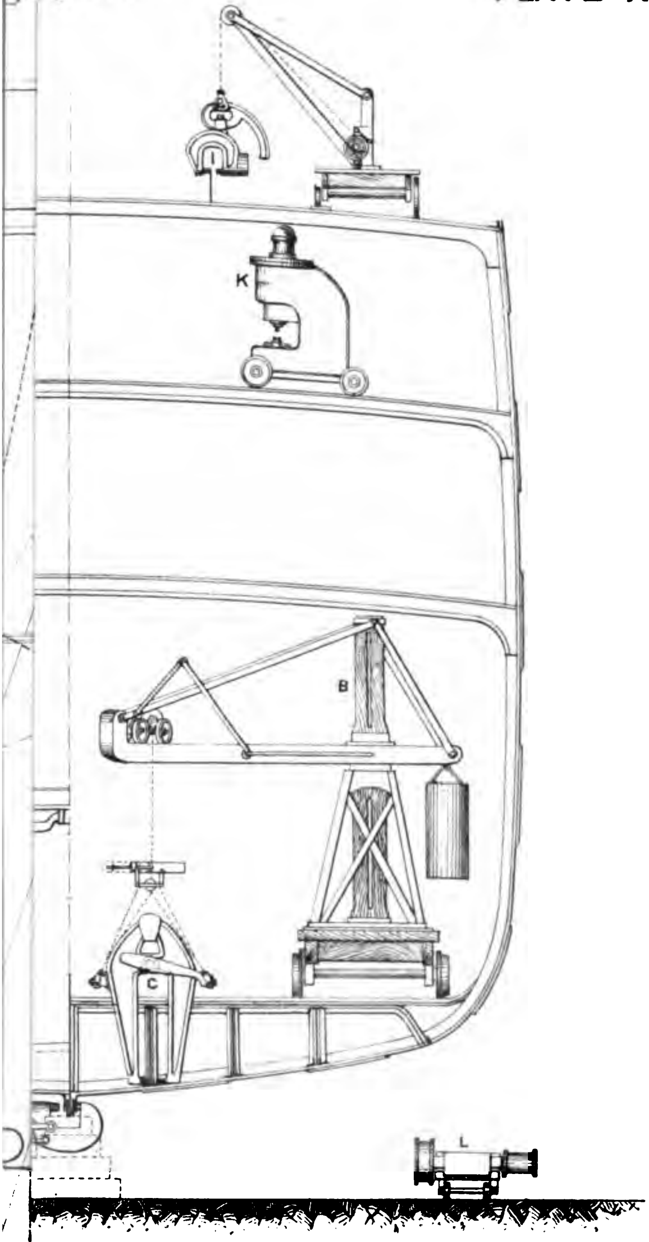
3/8" = 1 Foot.

PLATE XVII.



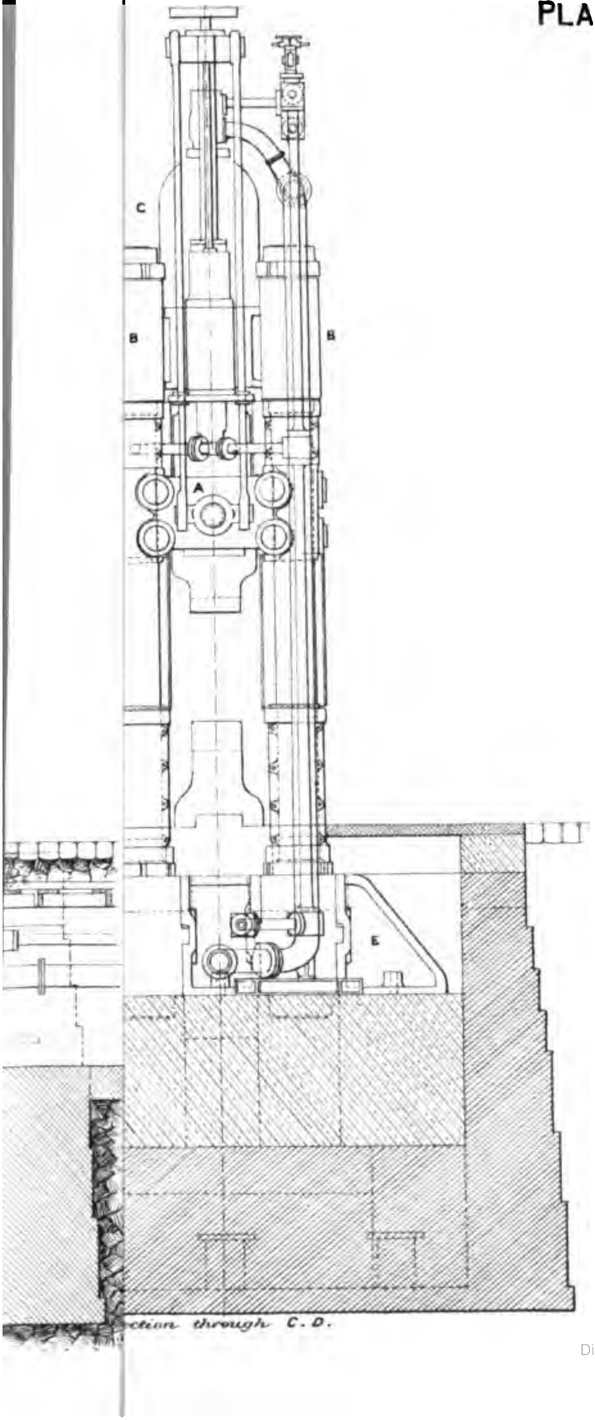
g ig. 2.

PLATE XVIII.



ch = 1 Foot.

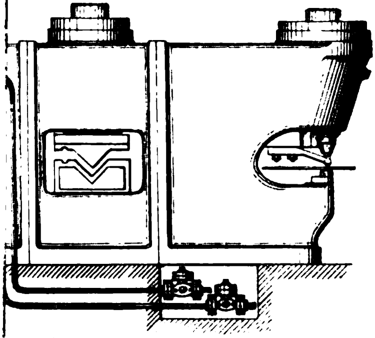
PLATE XIX.



ction through C. D.

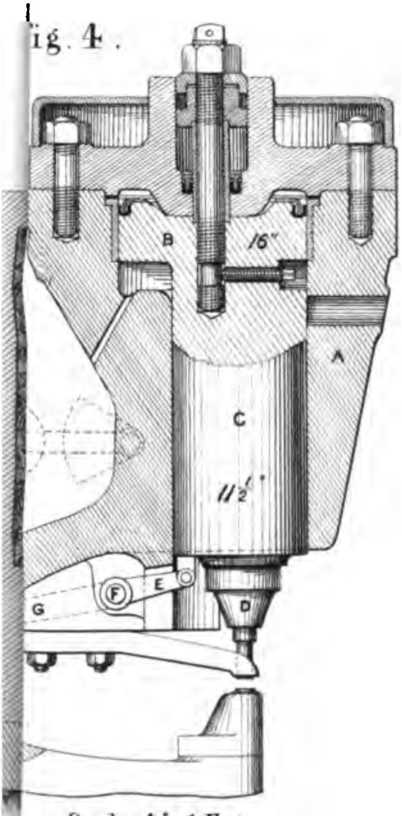
Fig. 5.

PLATE XX.



3/16" = 1 Foot

ig. 4.



Scale, ¼" = 1 Foot.

PUNCHING AND SHEARING MACHINE.

PLATE XXI.

FRANCIS.

Fig. 5.

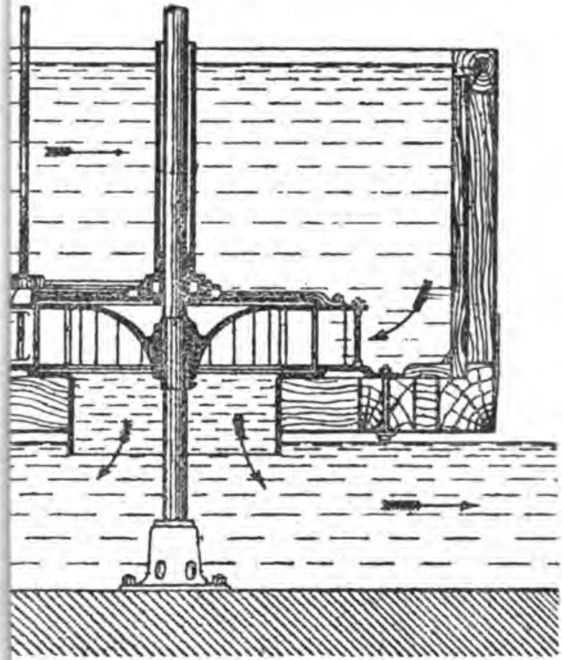


Fig. 6.

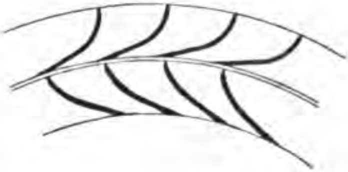


PLATE XXII.

Fig. 3.

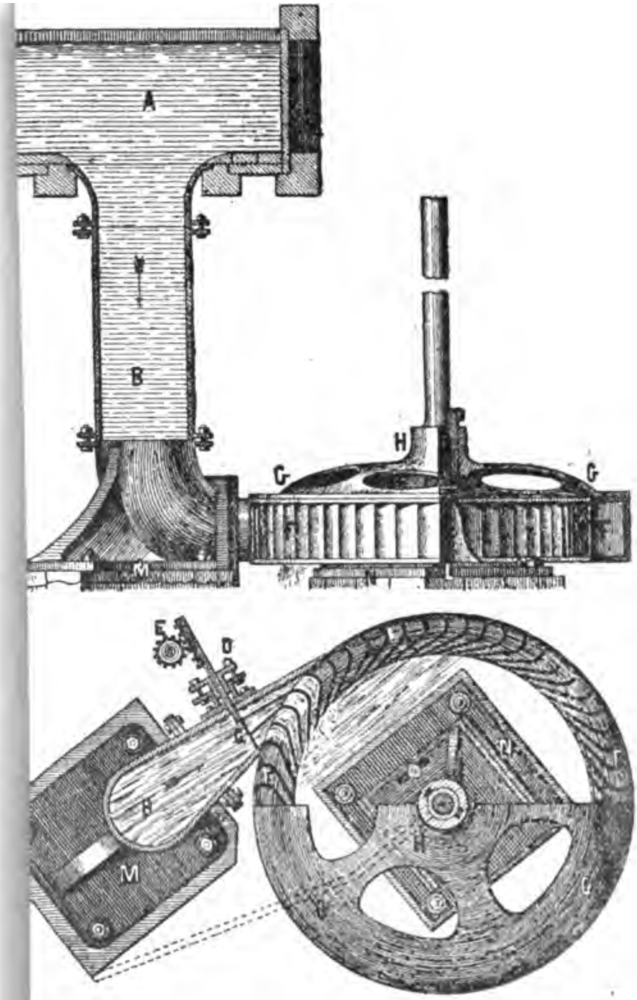
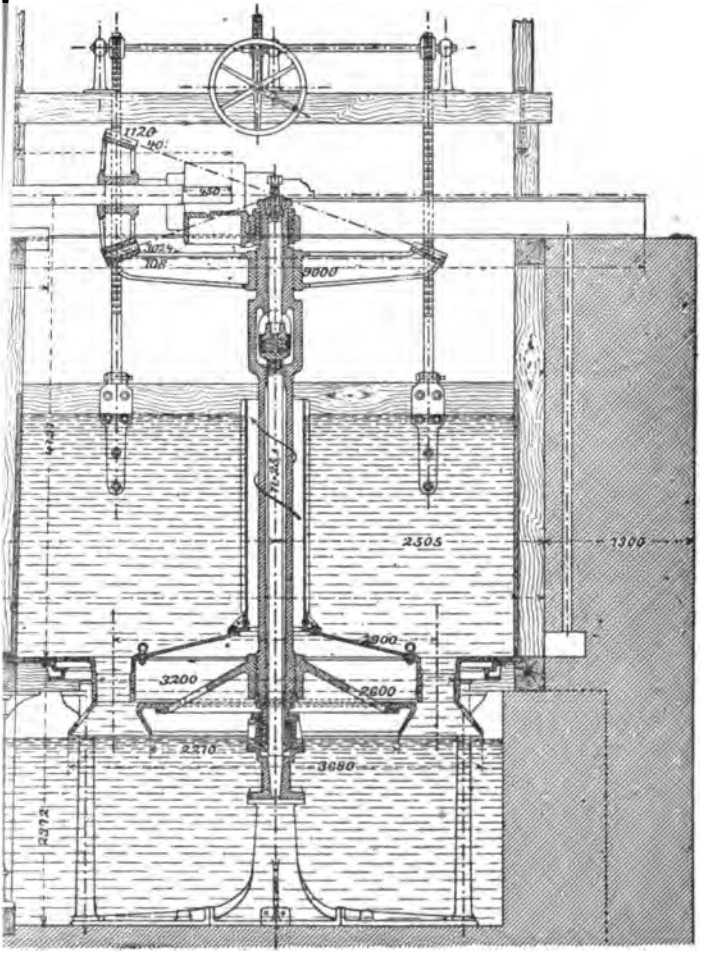
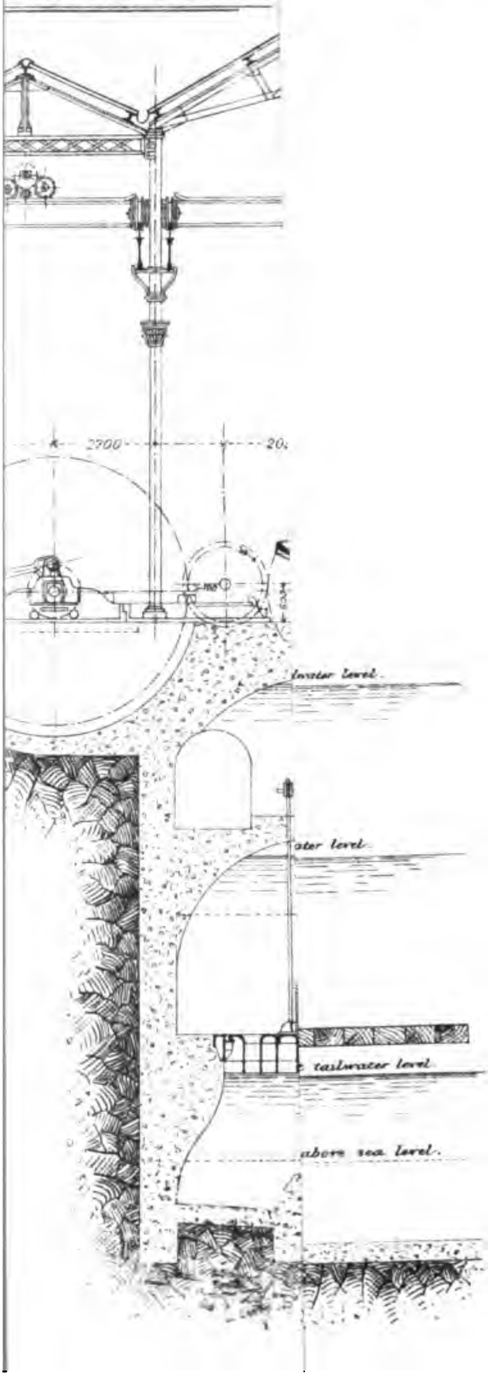


PLATE XXIII.



WATERWORKS.

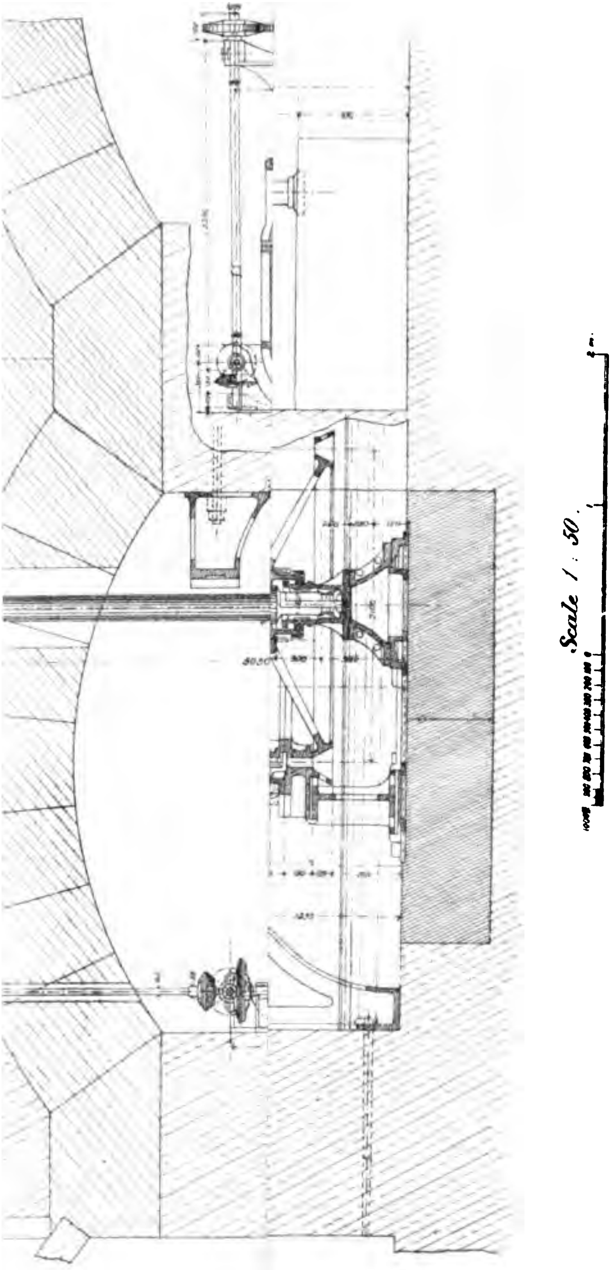
PLATE XXIV.



JLSE TURBINI

, SWITZERLAND.

PLATE XXV.



STALLATION AT

PLATE XXVI.

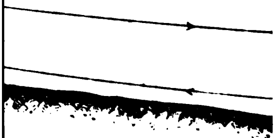


Fig. 3

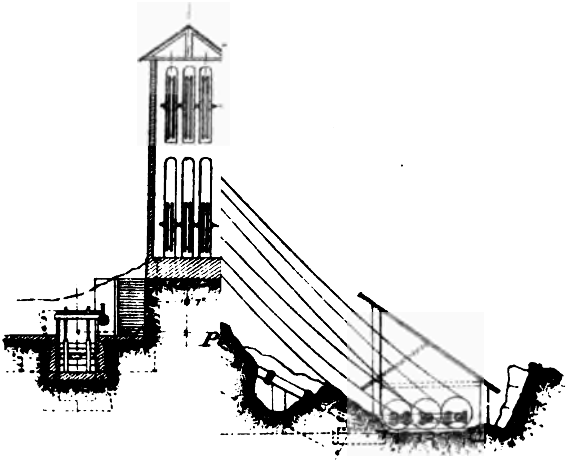


Fig. 2.

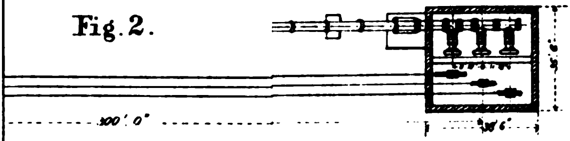


PLATE XXVII.

Fig. 2.

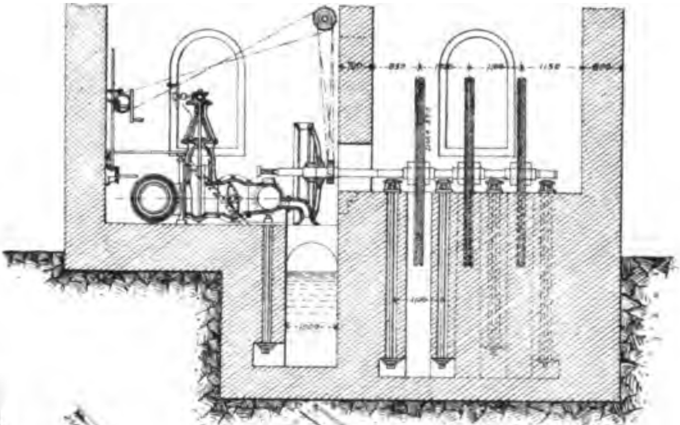
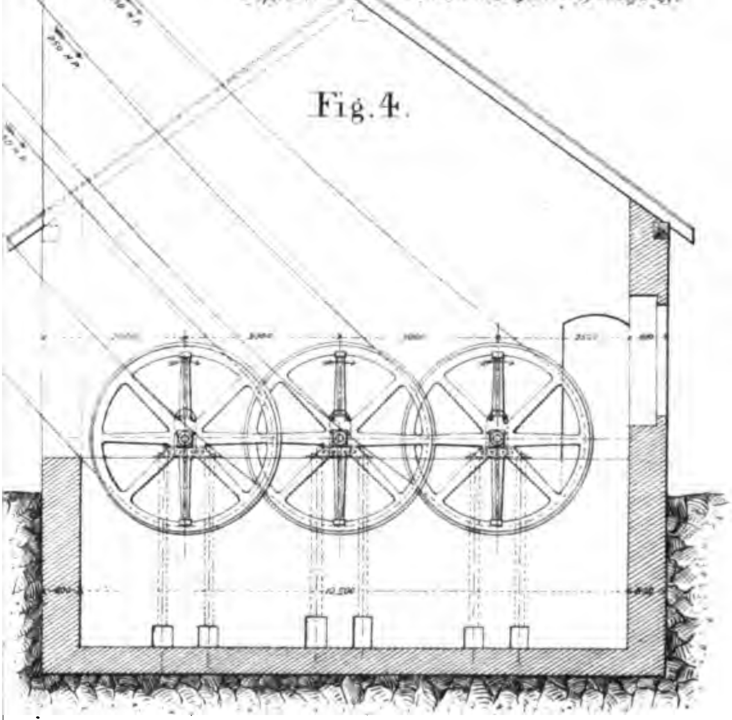


Fig. 4.



AT TERNI (I

PLATE XXVIII.

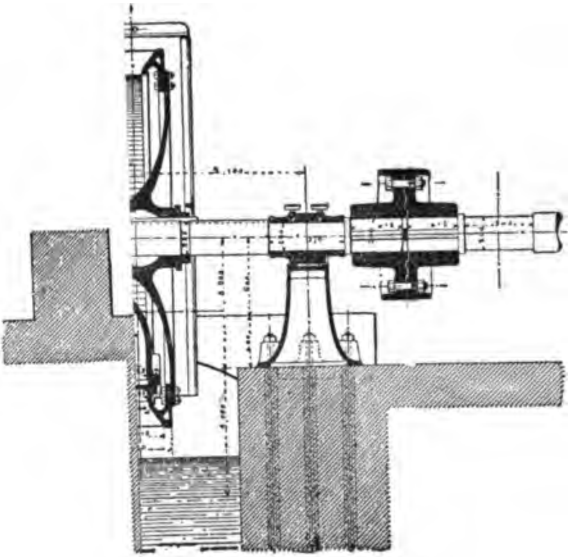


PLATE XXIX.

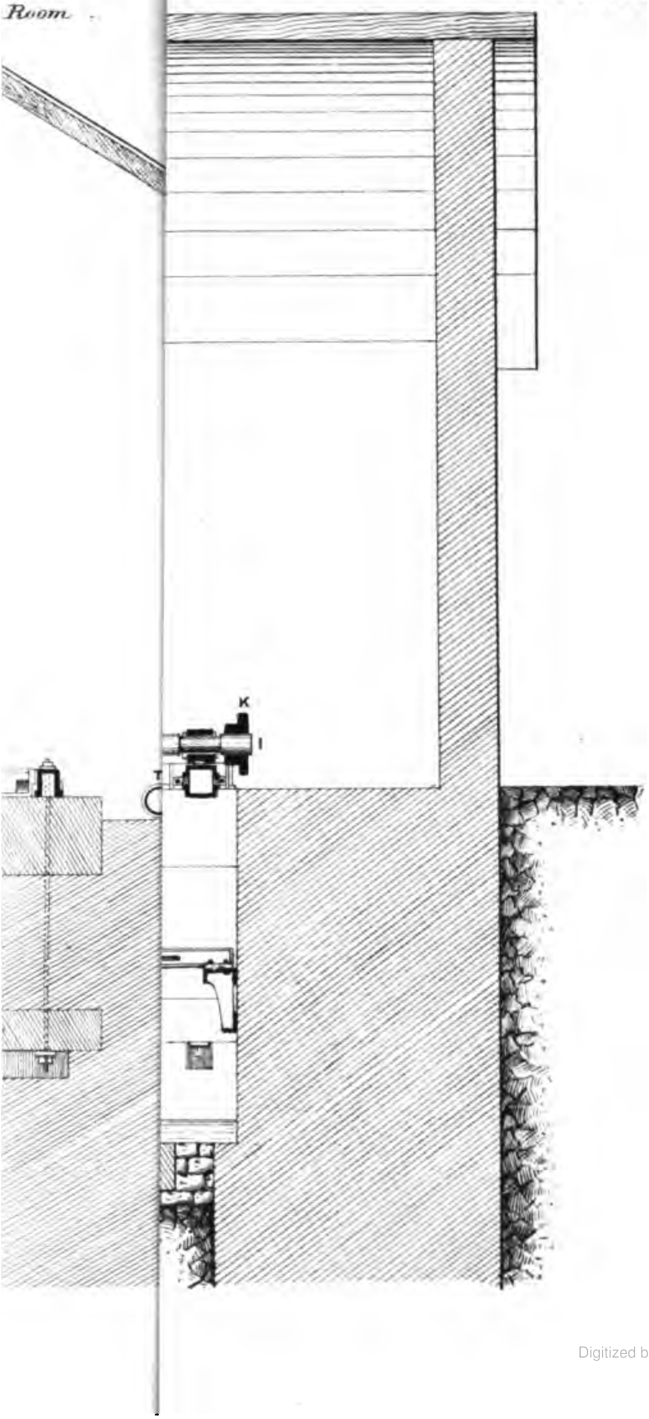
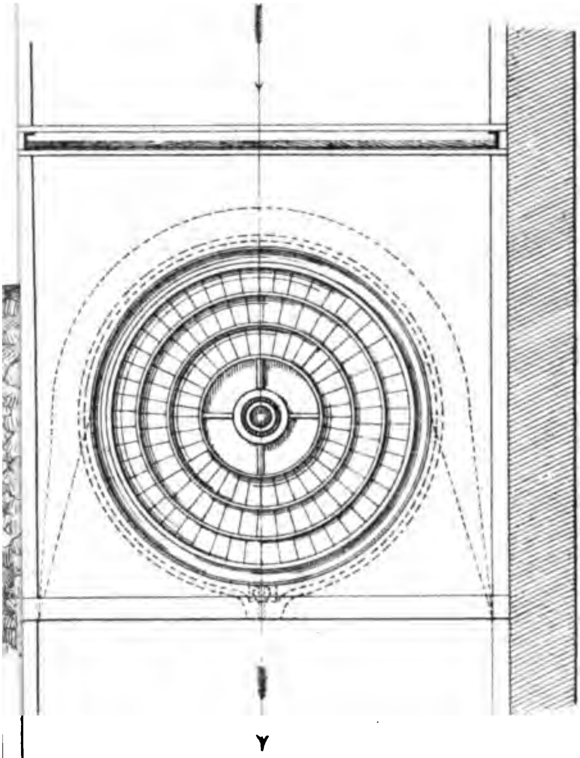


PLATE XXX.

Fig. 3.



PAPER VI.

LISTS OF
SERVICE ORDNANCE,

WITH DETAILS OF

NS, AMMUNITION, CARRIAGES, AND SLIDES.

1.—**Breech-Loading Ordnance (B.L. & R.B.L.).**
2.—**Muzzle-Loading Ordnance (R.M.L.).**
3.—**Quick-Firing and Machine Guns.**

Corrected to June, 1891.

1.—BREECH-LOAD

BREECH-LOAD

Ordnance.						Bore.		Chamber	
Nature.	Mark.	Material.	Weight.	Service.	Total Length in Ins.	Length in Inches.	Length in Calibres.	Diameter in Inches.	Length to base of Projectile.
16·25-inch	I	Steel	111 tons	S	524·0	487·5	30·0	21·125	84·5
13·5 ,, ..	I II	,, ... ,, ...	69 ,, 67 ,,	S	433·0	405·0	30·0	18·0	66·5
13·5 ,, ...	III IIIA,B,C,D,E,F IV	,, ... ,, ... ,, ...	67 ,,	S	433·0	405·0	30·0	18·0	66·5
12 ,, ...	I IA	Wrought-iron & steel ...	47 ,,	L	328·5	301·75	25·14	14·75	55·8
12 . ,, ...	III IV V	Steel	45 ,,	S	328·5	303·0	25·25	16·0	43·0
12 ,, ...	Vw	,, ...	46 ,,						
12 ...	VI VII	,, ...	46 ,,	L	328·5	303·0	25·25	16·0	48·0
10 ,, ...	I	,, ...	32 ,,	L	342·4	317·5	31·75	14·0	54·0
10 ,, ...	II III, IIIA, & IV	,, ...	29 ,,	L & S	342·4	320·0	32·0	14·0	54·0
9·2 ,, ...	I II	Wrought-iron & steel ...	22 ,, 21 ,,	L & S S	255·8	235·23	25·56	11·0	44·0
9·2 ,, ...	III V	Steel ... ... ,, ...	24 ,, 22 ,,	S	310·0	289·8	31·5	12·0	43·0
9·2 ,, ...	IV, IVA, & VIB VI, VIA, & VII	,, ... ,, ...	23 ,, 22 ,,	L L & S	310·0	289·8	31·5	12·0	43·0
8* ,, ...	III ,,	Steel unchase hooped... .. ,, chasehooped	13 ,, 14 ,,	S S	226·3 222·5	204·9	25·6	10·5	34·5
8 ,, ...	IV	Steel	15 ,,	S	254·5	236·9	29·61	10·5	38·0

ORDNANCE (B.L. AND R.B.L.).

ORDNANCE (B.L.).

Rifling.							Ballistic Effects.			Ordnance.		
System.	Mark.	Twist.	Length in Inches	Obturation.	No. of Grooves.	Venting.	Muzzle Velocity in Foot Seconds.	Muzzle Energy in Foot Tons.	Penetration of Wrought Iron at 1,000 Yds. in Ins.	Calibre in Inches.	Mark.	Weight.
Polygroove E.O.C.	I	From 1 in 130 at breech to 1 in 30 at 77·2 ins. from the muzzle; remainder uniform 1 in 30	397·2	Pad	78	Axial	2087	55237	32·0	16·25	I	111 tons
Polygroove (hook section)	I	From 1 in 120 at breech to 1 in 30 at 166·7 in. from the muzzle; remainder uniform 1 in 30	333·4	,,	54	,,	2016	35217	28·2	13·5 ,, ,, ,, ,,	I II III IIIA, B, C, D, E, F IV	69 ,, 67 ,, 67 ,,
,,	II	From 1 in 60 at breech to 1 in 30 at muzzle	333·4	,,	54	,,						
,,	I	From 1 in 04·5 at breech to 1 in 35 at 124·525 ins. from the muzzle; remainder uniform 1 in 35	2,1·45	,,	48	,,	1914	18187	20·4	12 ,, ,, ,, ,, ,, ,, ,,	I IA III IV V VW VI VII	47 ,, 45 ,, 46 ,,
,,	I	From 1 in 120 at breech to 1 in 35 at 126·275 ins. from the muzzle; remainder uniform 1 in 35	250·8	,,	48	,,						
,,	II	From 1 in 60 at breech to 1 in 30 at muzzle	250·8	,,	48	,,						
,,	II	From 1 in 60 at breech to 1 in 30 at muzzle.	259·68	,,	40	,,	20,0	14391	20·5	10 ,,	I II III, IIIA IV	32 ,, 29 ,,
,,			262·18	,,	40	,,						
,,	I	From 1 in 118·5 at breech to 1 in 35 at 81·12 ins. from the muzzle; remainder uniform 1 in 35	187·73	,,	37	,,	1809	8620	16·4	9·2 ,,	I II	32 ,,
,,	I	From 1 in 120 at breech to 1 in 30 at 120·4 ins from the muzzle; remainder uniform 1 in 30	243·4	,,	37	,,	2036	10915	18·3	,, ,, ,, ,,	III V IV, IVA, VIB VI, VII	24 ,, 22 ,, 23 ,, 22 ,,
,,	II	From 1 in 60 at breech to 1 in 30 at muzzle.	243·4	,,	37	,,						
,,	I	From 1 in 120 at breech to 1 in 35 at 67·7 ins. from the muzzle; remainder uniform 1 in 35	167·1	,,	32	,,	1953 2150	5[illegible]52 6729	13·4 14·9	8 ,, ,,	III ,, IV	13 ,, 14 ,, 15 ,,
,,	I	From 1 in 110 at breech to 1 in 35 at 99·7 ins. from the muzzle; remainder uniform 1 in 35	195·8	,,	32	,,	2200	7046	15·2	,,	VI	14 ,,

Ordnance.						Bore.		Chamber.		
Nature.	Mark.	Material.	Weight.	Service.	Total Length in Ins.	Length in Ins.	Length in Calibres.	Diameter in Inches.	Length to Base of Projectile.	Capacity in Cubic Inches.
8-inch	VI	Steel	14 tons	S	254·5	236·9	29·61	10·5	38 0	3350
8 ,, ...	VII	Wrt.-iron & steel unchasehooped	12 ,,	L	222·5	208·0	26·0	10·5	38·0	3240
8 ,, ...	VIIA	Wrt -iron & steel chasehooped	12 ,,	L	218·5	204·0	25·5	10·5	38·5	3292
6-inch 80-pr. ...	—	Wrt.-iron & steel unchasehooped	80 cwt.	S	165·6	156·2	26·03	7·5	27·4	1155
6 ,, ,, ...	—	Wrt.-iron & steel chasehooped	81 ,,	S	162·6	153·2	25·53	7·5	28·05	1185
6 ,, ...	II	Wrought-iron & steel	81 ,,	S	165·5	156·0	26·0	8·0	26·75	1364
6 ,, ...	†IIP	Wrt.-iron & steel chasehooped	89 ,,	L & S	153·5	144·0	24·0	8·0	26·75	1364
6 ,,	III ,,	Steel unchasehooped ,, chasehooped	89 ,, 5 tons	S	170·7	153·2	25·53	8·0	26·75	1364
6 ,,	†IV VI	Steel	5 ,,	L & S	173·5	156·0	26·0	8·0	26·75	1364
6 ,, ...	V	,, ...	5 ,,	L	195·3	183·5	30·58	8·0	31·75	1515
5 ,,	I †IP	,, ...	38 cwt.	S	139·5	125·35	25·07	5·75	19·05	504

ORDNANCE (B.L.).—(*Continued*)

Rifling.						Venting.	Ballistic Effects.			Ordnance.		
System.	Mark.	Twist.	Length in Inches	Obturation.	No. of Grooves.		Muzzle Velocity in Foot Seconds.	Muzzle Energy in Foot Tons.	Penetration of Wrought Iron at 1,000 Yds. in Ins.	Calibre in Inches.	Mark.	Weight.
Polygroove (hook section).	II	From 1 in 60 at breech to 1 in 30 at muzzle	125·8	Pad	32	Axial						
Polygroove E.O.C.	I	From 1 in 100 at breech to 1 in 40 at 10·82 ins. from the muzzle; remainder uniform 1 in 40	166·82	Cup	33	,,	2000	4992	12·8	8	VII	12 tons
,,	I	From 1 in 100 at breech to 1 in 40 at 6·82 ins. from the muzzle; remainder uniform 1 in 40	162·82	,,	33	,,				,,	VIIA	,,
,,	I	From 0 at breech to 1 in 40 at 7·02 ins. from the muzzle; remainder uniform 1 in 40	126·02	,,	28	,,	1880	1960	8·5	6 (80-pr.)	—	80 cwt.
,,	I	From 0 at breech to 1 in 40 at 4·02 ins. from the muzzle; remainder uniform 1 in 40	123·02	,,	28	,,				,,	—	81 ,,
Polygroove (hook section).	I	From 1 in 106·5 at breech to 1 in 35 at 65·125 ins. from the muzzle; remainder 1 in 35	126·875	,,	24	,,	1672	1938	8·8	6	II	81 ,,
,,	I	From 1 in 106·5 at breech to 1 in 35 at 53·125 ins. from the muzzle; remainder uniform 1 in 35	114·875	,,	24	,,				,,	IIP	89 ,,
,,	I	From 1 in 120 at breech to 1 in 35 at 58·9 ins. from the muzzle; remainder uniform 1 in 35	124·075	Pad	24	,,	1900 1870½	2663	10·5	,, ,, ,, ,,	III ,, IV VI	89 ,, 5 tons 5 ,,
,,	I	From 1 in 120 at breech to 1 in 35 at 61·75 ins. from the muzzle; remainder uniform 1 in 35	126·875	,,	24	,,						
	II	From 1 in 60 at breech to 1 in 30 at muzzle	126·875	,,	24	,,						
Polygroove E.O.C.	I	From 0 at breech to 1 in 30 at muzzle	149·75	Cup	28	,,	1920	2585	10·5	,,	V	5 ,,
Polygroove (hook section).	I	From 1 in 117 at breech to 1 in 30 at 52·4 ins. from the muzzle; remainder uniform 1 in 30	104·3	Pad	20	,,	1800	1123	6·9	5 ,,	I IP	38 cwt.

Ordnance.						**Bore.**		**Chamber.**		
Nature.	**Mark.**	**Material.**	**Weight.**	**Service.**	**Total Length in Ins.**	**Length in Ins.**	**Length in Calibres.**	**Diameter in Inches.**	**Length to base of Projectile.**	**Capacity in Cubic Inches.**
5-inch	II	Steel	38 cwt.	S	139·5	125·35	25·07	5·75	19·05	504
5 ,, ...	III IV V	Steel	40 ,,	L & S	139·15	25·0	25·0	5·75	19·05	504
4 ,, 13 cwt....	—	Wrought-iron & steel	13 ,,	S	66·0	59·25	14·81	4·5	8·125	126·5
4 ,, ...	I	Wrought-iron & steel	22 ,,	S	106·75	100·0	25·0	5·3	21·4	461
†4 ,, ...	II, IIP III	Steel	23 ,,	S	120·0	108·0	27·0	5·8	18·5	417
	IV V VI	Steel	26 ,,	L & S						
20-pr.	I	Steel (3·4-inch calibre)	12 ,,	L	107·5	98·5	29·0	3·9	15·3	168
†12-pr.	I	Steel (3-inch calibre)	7 ,,	L	92·35	84·0	28·0	3·625	11·0	117
32-pr. S.B.B.L.	I	Cast-iron (6·35-inch calibre).	42 ,,	L	97·6	86·0	13·5	6·55	4·1	136

B.L. guns of 6-inch calibre and upwards for future manufacture, or when repaired with new A tubes, or through-lined, will be rifled with a uniformly increasing twist, from 1 in 60 at the breech to 1 in 30 at the muzzle. This system is termed Mark II. rifling.

* 8-inch B.L. guns, Marks I and II, have been manufactured, but have been appropriated for proof of powder, and will not be issued to the service. Mark V gun has been designed but not manufactured.

† The addition P to a Mark indicates that the gun has been converted from the system of friction to the percussion-firing.

ORDNANCE (B.L.).—(*Continued*).

Rifling.						Venting.	Ballistic Effects.			Ordnance.		
System.	Mark.	Twist.	Length in Inches.	Obturation.	No. of Grooves.		Muzzle Velocity in Foot Seconds.	Muzzle Energy in Foot Tons.	Penetration of Wrought Iron at 1,000 Yds. in Inches.	Calibre in Inches.	Mark.	Weight.
Polygroove (hook section).	I	From 1 in 120 at breech to 1 in 25 at 52·15 in. from the muzzle; remainder uniform 1 in 25	104·3	Pad	20	Axial	1800	1123	6·9	6 ,, ,, ,,	II III IV V	83cwt. 40cwt.
,,	I	From 1 in 120 at breech to 1 in 25 at 51·8 ins. from the muzzle; remainder uniform 1 in 25	103·95	,,	20	,,						
,,	I	From 1 in 116 at breech to 1 in 35 at 11·39 ins. from the muzzle; remainder uniform 1 in 35	49·315	Cup	8	Rad'l C'per	1180	241	3·4	4	—	18cwt.
,,	I	From 1 in 120 at breech to 1 in 35 at 38·37 ins. from the muzzle; remainder uniform 1 in 35	76·87	,,	16	,,	1790	555	5·0	,,	I	22cwt.
,,	I II	From 1 in 120 at breech to 1 in 30 at 43·77 ins. from the muzzle; remainder uniform 1 in 30	87·77	Pad	16 24	Axial ,,	1900	626	5·4	,, ,, ,, ,, ,,	II, IIP III IV V VI	23cwt. 26cwt.
,,	—	From 1 in 120 at breech to 1 in 25 at 40·9 ins. from the muzzle; remainder uniform 1 in 25	81·8	,,	21	Rad'l Steel	1667	385	—	20-pr.	I	12cwt.
,,	I II	From 1 in 120 at breech to 1 in 28 at 35·8 ins. from the muzzle; remainder uniform 1 in 28	71·6	,,	12 18	,, ,,	1720	249	—	12-pr.	I	7cwt.
—	—	Smooth-bore ...	—	Cup	—	Rad'l C'per	—	—	—	32-pr., S.B.B.L.	I	42cwt.

‡ Some of the 6-inch Mark IV., of later manufacture, have Mark II. rifling.

§ With charges of Prism¹ Black and P².

‖ All 4-inch and 12-pr. B.L. guns of later manufacture will be rifled with six grooves per inch of calibre. The depth of the grooves in 4-inch guns will be reduced from 0·5 to 0·4-inch. Existing guns when "through-lined," or repaired with a new A tube, will have the modified rifling.

BREECH-LOADING

	Mark.	Material.	Service.	Total Length.	Bore.		Chamber.		
					Calibre.	Length in Calibres.	Diameter.	Length to base of Projectile.	Capacity.
				Ins.	Ins.		Ins.	Ins.	Cub. Ins.
7-inch 82 cwt. ...	—	Wrought-iron	L & S	120·0	7·0	14·21	7·20	16·0	620·0
,, 72 ,, ...	—	,, ...	L	118 0	7·0	14·21 or 13·93	7·20	14·25	552·9
40-pr. 35 cwt. ..	—	Wrought-iron and Steel ...	L & S	121·0	4·75	22·39	4·96	13·5	257·8
,, 32 ,, ...	—	,, ,, ...	,,	120 0					
20-pr. 16 ,, ...	—	,, ,, ...	L	96·0	3·75	22·36	3·94	12·0	143 0
,, 15 ,, ,, 13 ,,	—	,, ,, ...	S	66·125	3·75	14·43	3·94	11·0	131·0
12-pr. 8 cwt.... ...	—	Wrought-iron and Steel ...	L & S	72·0	3·0	20·458	3·20	8·5	66·0
9-pr. 6 ,, ...	—	Wrought-iron	,,	62·0	3·0	17·5	3·20	7·0	55·1
6-pr. 3 ,, ...	—	.. ..	,,	60·125	2·5	21·2	2·625	7·0	57·9

BREECH-LOADING

Ordnance.		Fuzes.				Tubes.	Charges—Weight in lbs.
		Time.		Percussion.			
Nature.	Mark.	Land Service.	Sea Service.	Land Service.	Sea Service.		Full.
16·25-inch	I	—	One middle time & percussion to e'ch shrapn'l shell, 20 % sp're A; Middle Sen've B	—	Direct action ABCD	Pt	960 S.B.Ca.
13·5 ,,	I II	—	,,	—	,,	,,	630c S.B.C.
13·5 ,,	III IIIA, B, C, D, E, F IV						
12 ,,	I IA						
12 ,,	III IV V	Middle Sensitive AB	,,	Directaction ABC	,,	,,	295c Prism'l Brown
12 ,, ...	Vw						
12 ,,	VI VII						

ORDNANCE (R.B.L.).

Rifling.			Vent.	Ballistic Effects with Full Charges.				
						Penetration of Wrought-Iron Armour Plate.		
System.	Twist in Calibres.	Length.	Position	Muzzle Velocity.	Muzzle Energy.	At 1000 Yards.	At 2000 Yards.	
		Ins.		Ft.-secs.	Ft.-tons	Ins.	Ins.	
Polygrooved ...	U. 1 in 37 cals. ...	83·1	*a*Trough vent-piece, striking cartridge at centre of base.	1100	847	5	4	7in. 82cwt.
,, ...	,, ,,	82·9		1100	847	5	4	,, 72 ,,
Polygrooved ..	U. 1 in 36½ cals. ...	92·5		1160	308	—	—	40-pr., 35-cwt.
								,, 32 ,,
,, ...	U. 1 in 38 ,, ...	71·6		1000	191	—	—	20-pr., 16 ,,
								,, 15 ,,
,, ...	,, ,,	42·75		1100	156	—	—	,, 13 ,,
Polygrooved ...	U. 1 in 38 cals. ...	52·5		1239	119	—	—	12-pr. 8 cwt.
,, ...	,, ,,	45·5		1055	63	—	—	9-pr. 6 ,,
,, ...	U. 1 in 30 ,,	45·6		1046	41	—	—	6-pr. 3 ,,

a 40-pr. side-closing has a copper radial vent 6·5 inches from end of bore inclined at an angle of 45 degrees to vertical plane of axis of gun on right side.

ORDNANCE (B.L.).

Charges—weight in lbs.		Projectiles—Weight in lbs.												Ordnance.		
		Common Shell.			Shrapnel Shell.			Shot.								
Reduced for Sea Service.	Saluting.	Mark.	Weight of shell Empty, Plugged.	Weight of Bursting Charge.	Mark.	Weight of Shell Empty, Plugged.	Weight of Bursting Charge.	Mark.	Palliser.	Mark.	Armour Piercing Forged Steel.	Mark.	Case.	Calibre in Inches.	Mark.	Weight.
720*b* & 480*c* S.B.C.	—	II IIIp	1607 $1619\frac{13}{16}$	187½ 179¼	Io	1792	8	I	1800	I	1800	I	1800	16·25	I	111 tons
														13·5	I	69 tons
														,,	II	67 tons
472½*d* and 315*e* S.B.C.	—	IIp	$1164\frac{1}{16}$	$85\frac{9}{16}$	Ip	$1243\frac{15}{16}$	5¼	I	1250	I	1250	I	1250	,,	III	67 tons
														,,	IIIA, B, C, D, E, F	
														,,	IV	
221½*d* Prism¹ Brown	—	II	$681\frac{14}{16}$	$31\frac{9}{16}$	II	$711\frac{5}{16}$	$1\frac{15}{16}$	II	714	I	714	I	714	12	I	47 tons
														,,	IA	
		IIIo	618½	95	IIIo	$710\frac{14}{16}$	2							,,	III	45 tons
														,,	IV	
														,,	V	
147½*e* Prism¹ Brown		Vp	$634\frac{11}{16}$	79	IV	$709\frac{11}{16}$	3							,,	Vw	46 tons
														,,	VI	
														,,	VII	

Ordnance.		Fuzes				Tubes	Charges—Weight in lbs.
Nature.	Mark.	Time.		Percussion.			Full.
		Land Service.	Sea Service.	Land Service.	Sea Service.		
10-inch ..	I	Middle Sensitive AB	One Middle Time & Percussion to each Shrapnel shell, 20 % spare A; Middle Sensitive B	Direct action ABC	Direct action ABCD	P	252c Prism^{l} Brown
10 ,,	II III, IIIA, & IV						
9·2 ,,	I II	,,	,,	,,	,,	P	144c Prism^{l} Brown
9·2 ,,	III V						166c Prism^{l} Brown
9·2 ,,	IV, IVA & VIB VI, VIA & VII						
8 ,,	III ,,	—	,,	—	,,	P	100c Prism^{l} Black
8 ,, ..	IV						118c Prism^{l} Brown
8 ,, ...	VI						
8 ,, ...	VII	*Time & Concussion.* *Medium w* & Time & Percussion Middle *q* Middle Sensitive B	—	Direct action ABC	—	M'ch'c'l & Electric	90c Prism^{l} Black
8 ,, ...	VIIA						
6-in. 80-pr.	—	—	One Middle Time & Percussion to each Shrapnel shell, 20 % spare A; Middle Sensitive B	—	Direct action ABCD	,,	34 S.P. ...
6 ,, ...	—						
6 ,, ...	II					V	36*dg* E.X.E., 34*e* P^{2}
6 ,, ...	IIP					P	
6 ,,	III ,,	*Time & Concussion.* *Medium w* & Time & Percussion, Middle *q* Middle Sensitive B	,,	Direct action ABC	,,	P	48*cgh*, 36*d* E.X.E., and 42 P^{2}
6 ,, ...	‡IV VI						48c E.X.E., 45*e* Prism^{l} Black, and 42c P^{2}
6 ,, ..	V						45*e* Prism^{l} Black Mark II, 42*e* P^{2}

ORDNANCE (B.L.).—(*Continued*).

Charges—Weight in lbs.		Projectiles—Weight in lbs.												Ordnance.		
		Common Shell.			Shrapnel Shell.			Shot.								
Reduced for Sea Service.	Saluting.	Mark.	Weight of Shell, Empty, Plugged.	Weight of Bursting Charge.	Mark.	Weight of Shell Empty, Plugged.	Weight of Bursting Charge.	Mark.	Palliser.	Mark	Armour Piercing Forged Steel.	Mark.	Case.	Calibre in Inches.	Mark.	Weight.
108*d* & 126*e* Prism[1] Brown	—	I*p*	$461\frac{5}{16}$	$37\frac{3}{4}$	I*p*	$497\frac{10}{16}$	$1\frac{9}{16}$	I	500	I	500	—	—	10 ,,	I II III, IIIA IV	82 tons 29 ,,
108*d* & 72*e* Prism[1] Brown 124½*d* & 83*e* Prism[1] Brown	— —	II III*o* V*p*	$361\frac{1}{16}$ $346\frac{1}{2}$ $348\frac{11}{16}$	$18\frac{1}{2}$ 33 $31\frac{14}{16}$	II III*o* IV*p*	$377\frac{13}{16}$ $377\frac{6}{16}$ 377	$1\frac{6}{16}$ $1\frac{1}{4}$ $2\frac{3}{16}$	IV	380	I	380	I	380	9·2 ,, ,, ,, ,, ,,	I II III V IV, IVA, VIB VI, VII	32 ,, 24 ,, 22 ,, 23 ,, 22 ,,
75*d* & 50*e* Prism[1] Black 88½*d* & 59*e* Prism[1] Brown	— —	II III*o* IV*o*	$195\frac{7}{16}$ $180\frac{1}{4}$ $190\frac{9}{16}$	$13\frac{1}{4}$ 29 $15\frac{1}{2}$	III IV*o* V*p*₁	$208\frac{2\cdot5}{16}$ $207\frac{3}{4}$ $208\frac{1}{4}$	$\frac{15\cdot5}{16}$ $1\frac{2}{16}$ $\frac{15}{16}$	II	210	I	210	I	210	8 ,, ,, ,,	III ,, IV VI	13 ,, 14 ,, 15 ,, 14 ,,
67½*dx* Prism[1] Black	—	I II*p*	168 $163\frac{3\cdot75}{16}$	15 $15\frac{13\cdot25}{16}$	I II*p*	179 $177\frac{12}{16}$	1 $\frac{15}{16}$	I II	177 180	—	*v*	I	180	,, ,,	VII VIIA	12 ,, 12 ,,
25 S.P. ...	7	II III*l*	$74\frac{9\cdot5}{16}$ $81\frac{10}{16}$	$6\frac{2}{16}$ $7\frac{3}{4}$	II*o*	79	$\frac{3}{16}$	II	$78\frac{7}{16}$	—	—	I	70	6 (80-pr.) ,, 6 ,,	— — II IIP	80 cwt. 81 ,, 81 ,, 89 ,,
24*eg* E.X.E. 17 P[2] 36*d* & 24*e* E.X.E. 21P[2] 36*d* & 24*e* E.X.E., 22½ Prism[1] Black and 21 P[2] —	7	III VI*p*	$91\frac{13}{16}$ $89\frac{4}{16}$	$7\frac{6}{16}$ $9\frac{13}{16}$	II IV*o*	$98\frac{1}{4}$ $98\frac{1}{2}$	$\frac{7}{16}$ $\frac{10\cdot5}{16}$	IV	10	I	100	I	100	,, ,, ,,	III ,, IV VI V	89 ,, 5 tons 5 ,, 5 ,,

Ordnance.		Fuzes.				Tubes.	Charges—(Weight in lbs.
		Time.		Percussion.			
Nature.	Mark.	Land Service.	Sea Service.	Land Service.	Sea Service.		Full.
5-inch ...	I IP II	*Time & Concussion Medium w A*, & Time & Percussion Middle*q*A Middle SensitiveB	One Middle Time & Percussion to each Shrapnel Shell, 20 % spare A; Middle SensitiveB	Direct action ABC	Direct action ABCD	M	16*e* S.P. ...
5 ,, ...	III IV V					P	
4 ,, 13 cwt....	—					Short or solid drawn	2 5/16 R.L.G.^{4} & 3¼ R.L.G.^{2}
4 ,, ...	I	Time & Percussion Short, ABC, & Time & Percussion Middle*r*A	,,	Small percussion ABC	,,	Short or solid drawn P	12 S.P. ...
4 ,, ...	II, IIP III IV V VI						
20-pr.	I	Time & Percussion Short, ABC, & Time & Percussion Middle AB*rs*	—	,,	—	Short or solid drawn	6 S.P.... ...
12-pr.	I	Time & Percussion Short, AB, & Time & Percussion Middle AB*rs*	—	,,	—	Short or solid drawn	4 S.P. R.L.G.^{2} *bjx*
32-pr. S.B.B.L.	I	—	—	—	—	Short or solid drawn	3 R.L.G.^{2} ...

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a In 8 cartridges.
b In 6 cartridges.
c In 4 cartridges.
d In 3 cartridges.
e In 2 cartridges.
g This charge of E.X.E. is not to be used with unchasehooped guns.
h For guns on "Vavasseur Broadside" and "Vavasseur Central Pivot" carriages only.

j For use with star shell.
l Double shell.
m Star shell also used.
n Practice only.
o Forged-steel.
p Cast-steel.
q 20 per cent. for use at short ranges.
r A percentage for use at long ranges.
s Middle sensitive for use with star shell.

ORDNANCE (B.L.)—(*Continued*).

Charges—Weight in lbs.		Projectiles—Weight in lbs.												Ordnance.		
		Common Shell.			Shrapnel Shell.			Shot.								
Reduced for Sea Service.	Saluting.	Mk.	Weight of Shell, empty, plugged.	Weight of Bursting Charge, Plugged.	Mark.	Weight of Shell, empty, Plugged.	Weight of Bursting Charge, Plugged.	Mark.	Palliser.	Mark.	Armour piercing forged steel.	Mark.	Case.	Calibre in Inches.	Mark.	Weight.
8 S.P. ...	3	III V*o*	$46\frac{5\cdot25}{16}$ $41\frac{13\cdot75}{16}$	$4\frac{1}{16}$ $6\frac{13}{16}$	III*o*	$49\frac{9}{16}$	$\frac{4\cdot5}{16}$	I	50	—	—	I	50	5 ,, ,, ,, ,, ,,	I I P II III IV V	38 cwt. 40 cwt.
—	—													4	—	13 cwt.
		III VI	$21\frac{12}{16}$ $21\frac{7\frac{1}{2}}{16}$	$1\frac{6}{16}$ $2\frac{14\cdot5}{16}$	III V*o*	24 $24\frac{1\cdot5}{16}$	$\frac{3}{16}$ $\frac{1\cdot5}{16}$	I	25	—	—	I	25			
—	4													,, ,, ,, ,, ,, ,,	I II, II P III IV V VI	22 cwt. 23 cwt. 26 cwt.
—	—	*m*	$17\frac{4}{16}$	$2\frac{1}{16}$	—	$19\frac{1}{8}$	$\frac{1}{16}$	—	—	—	—	—	20	20-pr.	I	12 cwt.
—	$1\frac{1}{4}$	I*o* I*n* *m*	$10\frac{10}{16}$ $11\frac{9\cdot5}{16}$	$1\frac{8\cdot5}{16}$ $\frac{10}{16}$	I*o* I*ny*	$11\frac{10\cdot25}{16}$ $11\frac{10\cdot25}{16}$	$\frac{\cdot75}{16}$ $\frac{\cdot75}{16}$	—	—	—	—	I	$12\frac{15}{16}$	12-pr.	I	7 cwt
—	—	—	—	—	—	—	—	—	—	—	—	I	$54\frac{1}{2}$	32-pr., S.B.B.L	I	42 cwt.

t Tubes with ball only. Lower natures of guns to use up existing stock of tubes without ball.

v A segment shell weighing 180lbs. is also used.

w Middle sensitive time will be used when the existing stock of time and concussion medium is finished. The latter are obsolete as regards future manufacture.

x For Land Service.

y The existing stock of these shell will be used up over *sea* ranges, but no more will be made.

A For present use.

B For future use.

C Except where otherwise stated, the earlier marks of fuzes are to be used first. Of the 15sec. fuze, Mark III only is to be used.

D A graze fuze in addition will be recommended when a satisfactory pattern is settled.

Ordnance. — Nature and Weight.	Mark.	Fuzes.					Charges.	
		Time.		Percussion.		Tubes.	Silk Cloth Cartridges.	
		L.S.	S.S.	L.S.	S.S.		Full.	Saluting Blank. R.L.G. or L.G.
							lb.	lb. oz.
7-in. 82 cwt. ...	—	15secs. with det. III, mid. sensitive*b*	15secs. with det. III mid. sensitive*b*	Dir. Act. I* II and Dir. Act. III*b*	—	Short or solid drawn with primer	11 R.L.G.[2]	7 0
,, 72 ,, ...	—	,,	—	,,	—	,,	10 ,,	7 0
40-pr. 35 cwt. ...	—	15secs. with det. III, mid. sensitive*b*	15-ecs. with det. III mid. sensitive*b*	R.L.	R.L. perc. and Dir. Act. I* II and III, Dir. Act III*b*.	Short or solid drawn with primer	5 R.L.G.[2]	3 0
,, 32 ,, ...	—							
20-pr. 16 ,, ...	—	E. time	—	B.L. plain	—	Short or solid drawn	2½ ,,	1 S
,, 15 ,, ,, 18 ,,	—	—	E. time	—	R.L. and Dir. Act. I* II and III for Com. B.L. plain for segment	,,	2½ ,,	1 8
12-pr. 8 cwt. ...	—	15secs. with det. III short T & P*b*, E time for segment	E. time	B.L. plain	B.L. plain	Short or solid drawn	1½ R.L.G.[2]	1 0
9 ,, 6 ,, ...	—	,,	,,	,,	,,	,,	1⅛ ,,	1 0
6 ,, 3	—	,,	,,	,,	,,	,,	¾ ,,	0 10

b For future use.

ORDNANCE (R.B.L.).

Projectiles.													
With Lead Coating.										Solid Shot.	Case Shot.		
Empty.						Bursting Charges.							
Common Shell.		Segment Shell.		Shrapnel Shell.		Common Shell.		Segment Shell.	Shrapnel Shell.				
						P.&F.G.	L.G.						
Mk.	lb. oz.	Mk.	lb. oz.	Mk.	lb. oz.	lb. oz.	lb. oz.	lb. oz. gr.	lb. oz.	lb. oz.	Mk.	lb. oz.	
II	83 0	I	98 $9\frac{11}{16}$	II	97 0	8 4	6 8	3 2 0	0 8	—	VI	68 2⅓	7-in., 82cwt.
—	93 0					10 5	7 10						
—	98 0	I	98 $9\frac{11}{16}$	II	97 0	10 5	7 10	3 2 0	0 8	—	VI	68 2⅓	,, 72 ,,
II	38 5	I	38 9¾	I	39 0	2 6	2 4	0 13 0	0 3	40 13⅓	II	31 8	40-pr., 35cwt.
													,, 32 ,,
II	20 8	I	19 10	—	—	—	1 2	700 grs. Shell, F.G.	—	20 9¼	III	20 5¼	20-pr., 16 ,,
III	20 8	I	19 10	—	—	1 5	1 2	700 ,,	—	20 9¼	—	—	,, 15 ,,
													,, 13 ,,
III	10 12	I	10 8	II	10 12	—	0 8	550 grs.	0 0¾	11 7	IV	11 8	12-pr. 8 cwt.
III	8 3	I	8 5	II	8 12	—	0 6	300 ,,	0 0¼	8 13	III	9 0	9 ,, 6 ,,
—	—	I	5 7	—	—	—	—	200 ,,	—	6 2	III	5 9	6 ,, 3 ,,

CARRIAGES. L. (B.L. AND R.B.L.).

Ordnance.		Material.	Nature of Carriage (giving Length of Recoil, Size of Port, &c., when necessary).	Elevation in Degrees.	Depression in Degrees.	Height of Axis of Trunnions in Inches.	Radius of Arc Traversing.	Radii of Racers.		Weight in cwt.	
Nature of Guns.	Weight.							Front.	Rear.	Carriage (without Gun).	Limbers (without Horses).
							ft. in.	ft. in.	ft. in.		
12-inch	47 tons 46 ,,	Steel	Upper or lower tier	7	4	35	19 4½	—	—	162¾	
10-inch	32 ,, 29 ,,	,,	Disappearing, Mark I	15	5	182·6	—	6 0	6 0	963	
			,, ,, II	15	5*	183	—	14 6	14 6	†1180	
			,, ,, III	16	7	256·25	—	9 9¼ inr. rdi.	10 2¾ out. rdi.	1072	
			Barbette ,, I	15	6¼	48	7·9	—	—	228½	
			,, ,, II	17	5	36	—	—	—	223	
			,, ,, III	15	5	48	169·75	—	—	—	
9·2-inch	22 ,, 23 ,,	,,	Disappearing ,, I	15	5	171·25	—	5 6	5 6	795¾	
			,, ,, II	15	5	186	—	13 6	13 6	†1092	
			Barbette ,, I	15	6½	48·5	—	—	—	226	
			,, ,, II	17	5	36	—	—	—	223	
8-inch	12 ,,	Iron	,, ,, I	12½	5	29·12	—	—	—	91¾	
6-inch	5 ,, 89 cwt.	Steel	Mark IV Gun, disappearing, Mark II and II.*	20	5	124·5	5 10	4 4	4 4	285½	
			Mark IV Gun, disappearing, Mark IV.	20	5	124·5	5 10	4 4	4 4	263	
			Mark V Gun, disappearing, Mark I.	15	5	124·5	5 10	5 0	5 0	305	
			Mark V Gun, disappearing, Mark III.	20	5	124·5	5 10	4 4	4 4	296½	
			Barbette, Mark I, V.C.P.... ...	15	15	14	—	—	—	—	
			,, ,, I	15	7	36	—	—	—	52	
5-inch	40 ,,	,,	Siege, 6ft. parapet	25	5	78	—	—	—	35	
4-inch	26 ,,	,,	,, ,, ...	25	5	78	—	—	—	29	
			,, Mark I jointed gun ...	17	8	43·5	—	—	—	—	
20-pr.	12 ,,	,,	Field ,, I	16	5	41	—	—	—	—	
12-pr.	7 ,,	,,	,, ,, I	16	8	39·5	—	—	—	11½	10¾
			,, ,, II	16	5	40	—	—	—	11·41	10¾

R.B.L.

Nature of Guns.	Weight.	Material.	Nature of Carriage.	Elevation in Degrees.	Depression in Degrees.	Height of Axis of Trunnions in Inches.	Radius of Arc Traversing.	Front.	Rear.	Carriage (without Gun).	Limbers (without Horses).
7-inch	82 cwt.	Iron	Moncrieff, with platform	15	5	125·75	—	C 6 10 D 8 11	C 6 10 D 4 8½	262¾	
		,,	Sliding, Medium, No 1, 4ft. 3in., 3ft. 6in., or 2ft 7in. parapet.	13½	8½	31·25	—	—	—	31½	
		,,	,, No. 2, 4ft. 3in. or 3ft. 6in. parapet.	13½	8½	37	—	—	—	23¾	
		,,	,, No. 3, 3ft. 6in. parapet	13½	8½	38·5	—	—	—	31½	
		Wood	,, No. 15, Casemate F ...	8	6	29·75	—	—	—	15	
		,,	,, No. 16, Dwarf F	22	6	36·5	—	—	—	15½	
	72 ,,	,,	,, No. 17, Dwarf F	19	6	36	—	—	—	15	
		,,	,, No. 18, converted Naval F	8½	6	29·75	—	—	—	14½	
40-pr.	35 ,, 32 ,,	Iron	Common, standing	24	9	41	—	—	—	17½	
			Siege 6ft. parapet, side closing	15	5	77·5	—	—	—	26½	13½
			Sliding, Medium, No. 4	8	15	20·5	—	—	—	21	
		Wood	Siege	11	8½	52	—	—	—	28	12½
			Sliding, Medium, No. 19, casemate F.	17½	6	31	—	—	—	10½	
			,, No. 20, Dwarf F	20	5	33	—	—	—	11½	
20-pr. ..	16 ,,	Iron	,, Medium, No. 5	10	15	16	—	—	—	—	
32-pr. S.B.B L.	42 ,,	,,	,, ,, No. 6	10	15	31·5	—	—	—	14½	

* But can go to 7½ degrees if emplacement admits.
CD Pivots.

† Inclusive of racer, holding-down bolts, and pivot.
F Elevation from bed, with screw removed.

SLIDES. L. (B.L., AND R.B.L.).

Ordnance.			Nature of Slide, whether Casemate, Dwarf, etc.	Height at Axis of Trunnions.	Radii of Racers.		Radius of Rack.	Radius of Traversing Arc.	Weight of Slide.	Nature of Carriage or Carriages used with Slide.
Nature of Guns.	Weight.	Material.			Front.	Rear.				
				in.	ft. in.	ft. in.	ft. in.	ft. in.	cwt.	
12-inch	47 tons	Iron	Lower tier	26·4	9 2	21 2	14 5	19 4¼	284¾	Upper or lower tier
	46 ,,		Upper tier	26·4	9 2	18 0	14 5	19 4¼	279¼	,, ,,
10-inch		Steel	Barbette, Mark I.	62	5 11·13	5 11·18	—	7 9	193¼	Barbette
	32 ,,		,, Mark I. D. pivot.	62	9 2·871	2 11·418	—	5 10	—	,,
	29 ,,		Barbette, Mark II.	77·06	10 10	10 10	—	11 10	911	,,
			,, Mark III.	64·843	10 10	10 10	—	11 10¼	—	,,
9·2-inch	22 ,,	Steel	,, Mark I...	61·4	5 11·38	5 11·38	—	7 9	209¼	,,
	23 ,,		,, Mark II.	77·06	10 10	10 10	—	11 10	911	,,
8-inch	12 ,,	Iron	,, E.O.C. ...	19	5 6·05	5 6·05	—	6 4½	77¼	,,
6-inch	5 ,,	Steel	,, Mark I. V.C.P.	22	2 3	2 3	—	—	—	,,
	89 cwt.		Barbette, Mark I.	20·25	3 9	3 9	—	—	50	,,

R.B.L.

Nature of Guns.	Weight.	Material.	Nature of Slide.	Height at Axis of Trunnions.	Front.	Rear.	Radius of Rack.	Radius of Traversing Arc.	Weight of Slide.	Nature of Carriage.
7-inch	82 cwt. 72 ,,	Iron	Trv'sing, Medium, No. 1, 4′ 8″ parapet, converted slide	34·25	5 0	14 0	—	—	50	Iron, sliding, medium, No. 1
			Trv'sing, Medium, No. 2, 3′ 6″ parapet, converted slide	25·25	5 0	14 0	—	—	50	,, ,,
			Trv'sing, Medium, No. 3, 2′ 7″ parapet, converted slide	14·25	5 0	14 0	—	—	50	,, ,,
		Wood	Trv'sing, Medium, No. 11, 4′ 8″ parapet	25·75	C 6 1 D 9 0 E 10 8¼ F 12 10	C 6 1 D 3 4¼ E 2 2 F 2 2	—	—	44	Iron, sliding, medium, Nos. 2, 10, & 13
			Trv'sing, Medium, No. 12, 3′ 6″ parapet	18·5	5 0	16 6	—	—	34¾	,, ,,
			Trv'sing, Medium, No. 13, 3′ 6″ para- 11ft. long)	17·75	5 0	14 0	—	—	3 ¾	Iron, sliding, medium, Nos. 3 & 11
			Trv'sing, Medium, No. 14, casemate and all S.B. guns	13·5	5 0	16 6	—	—	27	Wood, sliding, medium, Nos. 15, 19, 21, & 25
			Trv'sing, Medium, No. 15, converted slide	13·75††	—	—	—	—	18¾	Wood, sliding, medium, No. 18
			Trv'sing, Medium, No. 16, dwarf	26	C 6 1 D 9 0 E 4 8¼ F 12 10	C 6 1 D 3 4¼ E 2 2 F 2 2	—	—	37	Wood, sliding, medium, Nos. 16, 17, 22, & 23
		Ir'n	Trv'sing, Medium, No. 4	25·25	1 7½	8 7½	—	—	22	Iron, sliding, medium, No. 4
40-pr.	35 ,,	Wood	Trv'sing, Medium, No. 14, and S.B. guns, casemate	13·5	5 0	16 6	—	—	27	Wood, sliding, medium, Nos. 15, 19, 21, & 25
	32 ,,		Trv'sing, Medium, No. 17, and S.B. guns, dwarf	25·5	5 0	16 6	—	—	32¾	Wood, sliding, medium, Nos. 20, 23, & 24
20-pr.	16 ,,	Iron	Trv'sing, Medium, No. 5	28·5	1 5	6 10	—	—	—	Iron, sliding, medium, No. 5
32-pr. S.B., B.L.	42 ,,	Iron	Trv'sing, Medium, No. 6	11·5	1 6	6 10	—	—	13	Iron, sliding, medium, No. 6

†† From ground platform.

C, D, E, F pivots.

CARRIAGES, NAVAL (B.L.).

Ordnance.	Material.	Nature.	Elevation in Degrees.	Depression in Degrees.	Height at Axis of Trunnions, ins.	Weight of Carriage without Gun, cwt.
10-inch.	Steel	C Pivot, Vavasseur, Mk. I.	14	5	40*	72
9·2-inch.	Steel	Vavasseur, Broadside, Mk. I.	10	3½	34·75*	55¼
		" C Pivot, Mk. I.	15	5	52*	118¼
		" " Mk. II.	12	5	40·73*	49¾
		" " Mk. III.	15	5	39*	44
8-inch.	Steel	Vavasseur, Broadside, Mk. I.	10	4½	39	45¼
		" C Pivot, Mk. I.	15	5	35·75*	36
6-inch.	Steel	Armstrong, Mk. I.	13	8	34·5	23¼
		" Mk. I. modified	13	7	37·4	—
		Vavasseur, Broadside, Mk. I.	16	6	40	23¼
		" " Mk. II.	16	7	40	25½
		" " Mk. III.	16	12	40	20¾
		" C Pivot, Mk. I.	20	7	44	22¼
		" " Mk. II.	15	7	42·25	17½
40-pr.	Iron and Metal	C Pivot, Albini	10	15	42·98	57½
6-inch.	Iron and Metal	C Pivot, Albini	12½	15	47·75	—
5-inch.	Metal	Vavasseur, Broadside, Mk. I.	15	7½	37·15	11½
		" " Mk. II.	22	7½	37·15	11½
		" C Pivot, Mk. I.	20	7½	38·5	12
4-inch.	Metal	Vavasseur, Broadside, Mk. I.	18½	28	29·75	6
		" " Mk. II.	20	30	30·75	7
		" C Pivot, Mk. I.	20	14	48†	8¼
		Cradles, Naval, B.L.				
4·7-inch.	Metal	Quick-firing, Upper Deck, Mk. I.... ...	20	7	45	12½
		" " Mk. I.* ...	20	7	45	13¼
		" Between Decks, Mk. I. ...	15	7	43	13¼
		" General, Mk. II.	20	7	45	13¼
		Stands and Mountings.				
6-pr.	Iron	Hotchkiss, Quick-firing, Mk. I.	17	25	43	7
		Nordenfelt, " Mk. I.	18	25	43·43	7
		" " Mk. II.	18	25	43·43	7
3-pr.	Iron	Hotchkiss, " Mk. I.	25	35	43	7½‡
	Steel and Metal	Elswick pattern, Recoil, Mk. I.	25	30§	15·5‖	5¶

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* From platform to axis of trunnion.
† Eight shields for "Rattler" class were made, suitable for a height of 42·5 inch from deck.
‡ The pivot is included in this weight.
§ With shield on the depression is limited to 9 degrees.
‖ This height is from bottom of holding-down ring to axis of trunnion.
¶ Total weight of mounting, without shield.

SLIDES, NAVAL (B.L.).

Material.	Nature.	Radii of Racers.		Radius of Rack.	Weight of Slide.	Reference to Paragraph for List of Changes in War Material.	
		Front. ft.††ins.	Rear. ft.††ins.	ft.‡‡ins.	cwt.	Carriage No.	Slide No.
Steel	C Pivot, Vavasseur, Mk. I.	4 4·5	4 4·5	3 6·275	158¼	—	—
Steel	Vavasseur, Broadside, Mk. I.	6 6·5	11 9	11 4·4	107½	—	—
	,, C Pivot, Mk. I.	4 3·75	4 3·75	2 0·545	118	—	—
	,, ,, Mk. II.	4 3·5	4 3·5	3 6·275	227½	—	—
	,, ,, Mk. III.	4 4·5	4 4·5	3 6·275	253½	—	—
Steel	Vavasseur, Broadside, Mk. I.... ...	5 11	13 8	8 0	47	—	—
	,, C Pivot, Mk. I.	3 7·75	3 7·75	2 9·1	77¾	—	—
Steel	Armstrong, Mk. I.	30·4	10 1·43	11 1·9. 3	34¼	4563	4563
	,, Mk. I., modified	3 0·4	10 1·43	11 1·933	—	—	—
	Vavasseur, Broadside, Mk. I., 3ft. Pivot Bar	5 0	9 9·27	9 0·31	37¾	5104	4677
	Vavasseur, Broadside, Mk. I., 3ft. 6in. Pivot Bar	5 5·8	10 3·2	9 6·12	87¾	—	4677
	Vavasseur, Broadside, Mk. I., 3ft. 9in. Pivot Bar	5 8·6	10 6·16	9 9·03	37¾	—	4677
	Vavasseur, Broadside, Mk. II. ...	3 10·78	8 2·885	4 9	41¾	—	—
	,, ,, Mk. III. ...	3 10·78	8 2·885	4 9	41¾	—	—
	,, C Pivot, Mk. I.	2 10·5	2 10·5	1 6·6	45¾	—	—
	,, ,, Mk. II.	2 3·25	2 3·25	1 4·154	57	—	—
Steel	C Pivot, Albini	—	—	—	45½**	—	—
Steel	C Pivot, Albini	—	—	—	—	—	—
Steel	Vavasseur, Broadside, Mk. I.... ...	—	6 4·5	6 11·25	27¾	4564	4564
	,, ,, Mk. II. ..	—	6 4·5	3 6·5	26	—	—
	,, C Pivot, Mk. I.	2 0	2 0	0 10·1	19¼	5105	5105
Steel	Vavasseur, Broadside, Mk. I.... ...	—	4 3·5	—	11½	4168	4168
	,, ,, Mk. II. ...	5 6	3 8	—	22	4648	4493
	,, C Pivot, Mk. I.	1 7	1 7	0 8·683	13¼	—	—
	Carriages, Naval (B.L.).						
Steel	Quick-firing, Upper Deck, Mk. I. ...	1 6·875	1 6·875	0 11·46	29¾	6010	—
	,, ,, Mk. I.*...	1 6·875	1 6·875	0 11·46	29¾	—	—
	,, Between Decks, Mk. I.	—	—	—	38½	—	—
	,, General, Mk. II. ...	1 6·785	1 6·875	0 11·46	27	—	—
—	— —	—	—	—	—	5008	—
—	— —	—	—	—	—	5009	—
—	— —	—	—	—	—	5161	—
—	— —	—	—	—	—	5134	—

** Weight of kerb and clip plates.

†† The "radii of front and rear racers" includes the radii of the pivot plates of C.P. mountings.

‡‡ The "radii of rack racers" includes the radii of the training worm wheels of C.P. mountings.

2.—MUZZLE-LOADING

ORDNANCE. — Guns.	Mark.	Material.	Service.	Total Length.	Bore.		Chamber.		
					Calibre.	Length in Calibres.	Diameter in Inches.	Length in Inches.	Capacity in Cubic Inches.
				ins.	ins.				
17·72-in. (100 tons) ...	I	Wrought-iron, with steel tube	L	391·85	17·72	20·48	19·7	59·72	17049
16-in. (80 tons)... ...	I	,, ,,	L & S	321·0	16·0	18·0	18·0	59·6	14600
12·5-in. (38 tons) ...	I	,, ,,	L & S	230·0	12·5	15·84	Unchamb'd	...	...
12·5-in. (38 tons) ...	II	,, ,,	L & S	222·8	12·5	15·84	14·0	41·125	6000
12-in., 35-ton	I	,, ,,	L & S	195·0	12·0	13·54	Unchamb'd	...	...
12-in., 25-ton	II	,, ,,	L & S	182·5	12·0	12·09	,,	...	...
11-in. (25 tons)... ...	II	,, ,,	L & S	180·0	11·0	13·18	,,	...	...
10·4-in. (28 tons) ...	I	,, ,,	L	289·0	10·4	26·00	12·5	56	6666
10-in. (18 tons)... ...	II	,, ,,	L & S	180·0	10·0	14·55	Unchamb'd	...	...
9-in. (12 tons)	V	,, ,,	L & S	156·0	9·0	13·89	,,	...	...
9-in. (12 tons)	VI	,, ,,	L	156·0	9·0	13·89	,,	...	...
8-in. (9 tons)	I III	,, ,, ,, ,,	L L & S	144·0	8·0	14·75	,,	...	...
7-in., 7-ton	IV	,, ,,	L	148·0	7·0	18·0	,,	...	...
7-in., 6½-ton	III	,, ,,	L & S	133·0	7·0	15·86	,,	...	...
7-in., 90-cwt.	I	,, ,,	S	131·0	7·0	15·86	,,	...	...
6·6-in. (70 cwt.) ...	I	,, ,,	L	118·0	6·6	14·78	6·8	21·0	707
80-pr. (convd. 5 tons)	I	Cast-iron, W.I. tube	L	136·55	6·29	18·004	Unchamb'd	...	...
64-pr., 64-cwt.	I II	W.I., steel or W.I. tube	L	119·5 120·0	6·3	15·47	,,	...	...
64-pr., 64-cwt.	III	W.I., steel tube ...	L & S	118·0	6·3	15·47	,,	...	...
64-pr.(convd.)71-cwt.	I	Cast-iron, W.I. tube	L & S	132·72	6·29	16·42	,,	...	...
64-pr.(convd.)58-cwt.	I	,, ,,	L	127·45	6·29	17·24	,,	...	...
40-pr. (34 cwt.) ...	I	W.I., steel tube ...	L	100·5	4·75	18·0	,,	...	..
40-pr. (35 cwt.) ...	II	,, ,, ...	L	120·0	4·75	22·0	,,	...	...
25-pr. (18 cwt.) ...	I	,, ,, ...	L	98·0	4·00	22·0	,,	...	...
16-pr. (12 cwt.) ...	I	,, ,, ...	L	78·0	3·6	19·0	,,	...	...
15-pr. jointed (422lb.)	I	Steel	L	70·5	3·3	20·0	3·2	8·37	68
13-pr. (8 cwt.)	I	W.I., steel tube ...	L	92·0	3·0	23·0	3·15	14·13	110·3
9-pr., 8-cwt.	I & II	,, ,, ...	L & S	72·0	3·0	21·17	Unchamb'd	...	...
9-pr., 6-cwt.	I	,, ,, ...	S	61·0	3·0	17·67	,,	...	...
9-pr., 6-cwt.	II & III IV	,, ,, ... Steel	L & S S	74·5 74·875	3·0	22·0	,,	...	...

ORDNANCE (R.M.L.).

Rifling.			Vent.	Ballistic Effects with Full Charges.				ORDNANCE.
				Muzzle Velocity in foot-seconds.	Muzzle Energy in foot-tons.	Penetration of Wrought-Iron Armour Plate.		
System.	Twist in Calibres.	Length.	Position Ins. from end of Bore.			At 1000 yds.	At 2000 yds.	Guns.
Polygroove (plain section)	I. 1 in 150 to 1 in 50, at 2·88-in. from muzzle, remainder U.	302·88	Axial	1548	33233	23	21	17·72-in. (100 tons)
,, ...	I. 0 to 1 in 50 cals.	227·4	Axial	1540	29806	23	22	16-in. (80 tons)
Woolwich	I. 0 to 1 in 35 cals.	170·5	12·0	1442 (L.S.)	11823	16	15	12·5-in. (38 tons)
,, ...	I. 1 in 438 to 1 in 35 cals. ...	156·87	Axial	1575 (S.S.)	14140	18	16	12·5-in. (38 tons)
,, ...	I. 0 to 1 in 35 cals.	135·0	12·0	1390	9563	15	13	12-in., 35-ton
,, ...	I. 1 in 100 to 1 in 50 cals. ...	127·0	9·8	1292	7124	12	11	12-in., 25-ton
,, ...	I. 0 to 1 in 35 cals.	119·0	10·0	1360	6510	13	12	11-in. (25 tons)
Polygroove (plain section)	I. 1 in 200 to 1 in 40 cals. ...	213·4	Axial	1810	10400	17	15	10·4-in. (28 tons)
Woolwich	I. 1 in 100 to 1 in 40 cals. ...	118·0	11·0	1379	5406	12	10	10-in. (18 tons)
,, ...	I. 0 to 1 in 45 cals.	104·0	9·7	1440	3695	10	9	9-in. (12 tons)
Polygroove (plain section)	I. 1 in 100 to 1 in 35 at 22·5-in. from muzzle, remainder U.	98·0	2·5	...	...	...	...	9-in. (12 tons)
Woolwich	I. 0 to 1 in 40 cals.	102·0 99·5	9·2	1390	2391	8	7	8-in. (9 tons)
,, ...	U. 1 in 35 cals.	110·5	8·6	1561	1943	8	6	7-in., 7-ton.
,, ...	U. 1 in 35 cals.	95·535	8·6	1525	1854	8	6	7-in., 6½-ton.
,, ...	U. 1 in 35 cals.	95·535	8·6	1325	1400	7	6	7-in., 90-cwt.
Polygroove (plain section)	I. 1 in 100 to 1 in 35 at 13·2-in. from muzzle, remainder U.	76·5	5·2	1416	1398	...	...	6·6-in. (70 cwt.)
Woolwich	U. 1 in 40 cals.	106·25	1·85	1230	944	...	...	80-pr. (convd. 5 tons)
				1553	1337	(20 lb.	charge)	
Shunt or plain ...	U. 1 in 40 cals.	90·5	5·2 or 6·1 5·2	1125	588	...	...	64-pr., 64-cwt.
Plain	U. 1 in 40 cals.	90·5	5·2	1390	897	..	...	64-pr., 64-cwt.
,, ...	U. 1 in 40 cals.	96·27	1·7	1260	737	...	...	64-pr. (convd.), 71-cwt.
,, ...	U. 1 in 40 cals.	101·45	1·8	1260	7 7	...	...	64-pr. (convd.), 58-cwt.
Woolwich	U. 1 in 35 cals.	71·5	0·6	1340	473	...	...	40-pr. (34-cwt.)
,, ...	U. 1 in 35 cals.	90·5	1·0	1425	555	...	...	40-pr. (35 cwt.)
,, ...	U. 1 in 35 cals.	78·0	1·0	1350	322	...	..	25-pr. (18 cwt.)
French, modified	U. 1 in 30 cals.	58·04	0·6	1355	203	...	..	16-pr. (12 cwt.)
Polygroove (plain section)	I. 1 in 60 to 1 in 20 at 15·13-in. from muzzle, remainder U.	57·13	4·8	1040	112	...	...	15-pr., jointed (422 lb.)
,, ...	I. 1 in 100 to 1 in 30 at 9-in. from muzzle, remainder U.	69·0	7·0	1595	229	...	...	13-pr. (8 cwt.)
French, modified	U. 1 in 30 cals.	59·8	0·6	1330	110	...	...	9-pr., 8-cwt.
,, ...	U. 1 in 30 cals.	49·3	0·6	1250	97	...	...	9-pr., 6-cwt.
,, ...	U. 1 in 30 cals.	62·3	0·6	1390	121	...	...	9-pr., 6-cwt.

ORDNANCE. Guns.	Mark.	Material.	Service.	Total Length.	Bore.		Chamber.		
					Calibre.	Length in Calibres.	Diameter in Inches.	Length in Inches.	Capacity in Cubic Inches.
2·5-in.(400 lb. jointed)	I	Steel	L	70·45	2·5	26·6	2·56	11·07	54
7-pr , bronze, 200-lb.	II	Bronze	S	38·125	3·0	10·7	Unchamb'd	...	...
7-pr., 200-lb.	IV	Steel	L & S	41·0	3·0	12·0	,,	...	...
7-pr., 150-lb.	III	,, ...	L	29·125	3·0	8·0	,,	...	...
HOWITZERS.									
8-in., 70-cwt.	I	W.I., steel tube	L	113·0	8·0	12·0	,,	...	...
	II	Steel							
8-in., 46-cwt.	I	W.I., steel tube ...	L	64·0	8·0	6·0	,,	...	...
6·6-in., 36-cwt. ...	I	,, ,,	L	90·7	6·6	12·0	,,	...	...
	II	Steel							
6·3-in., 18-cwt.... ...	I	W.I., steel tube ...	L	56·0	6·3	7·14	,,	...	...
4-in., jointed, 600-lb.	I	Steel	L	57·45	4·0	13·0	,,	...	...

ORDNANCE (R.M.L.).—(*Continued*).

Rifling.			Vent.	Ballistic Effects with Full Charges.				ORDNANCE.
						Penetration of Wrought-Iron Armour Plate.		—
System.	Twist in Calibres.	Length.	Position. Ins. from end of Bore.	Muzzle Velocity in foot-seconds.	Muzzle Energy in foot-tons.	At 1000 yds.	At 2000 yds.	Guns.
Polygroove (plain section)	I. 1 in 80 to 1 in 30 at 3·53-in. from muzzle, remainder U.	54·73	5·25	1440	100	...	...	2·5-in. (400 lb. jointed)
French	U. 1 in 20 cals....	29·5	·75	914	40	...	...	7-pr., 200-lb.
,, ...	U. 1 in 20 cals....	34·0	1·0	950	47	...	...	7-pr., 200-lb.
,, ...	U. 1 in 20 cals....	22·0	1·0	673	23	...	...	7-pr., 150-lb.
								HOWITZERS.
Polygroove (plain section)	I. 1 in 90 to 1 in 35 cals. ...	88·0	2·0	956	1150	...	...	8-in., 70-cwt.
Woolwich	U. 1 in 16 cals....	35·5	1·75	697	608	...	...	8-in., 46-cwt.
Polygroove (plain section)	I. 1 in 94 to 1 in 35 cals. ...	74·125	1·5	839	488	...	...	6·6-in., 36-cwt.
,, ...	I. 1 in 100 to 1 in 35 cals. ...	39·7	1·125	751	285	...	...	6·3-in., 18-cwt.
,, ...	I. 0 to 1 in 25 at 8·3-in. from muzzle, remainder U.	47·0	1·0	835	97	...	...	4-in., 600-lb.

CE (R.M.L.).—*Continued.*

Studless Projectiles.												Tubes.	Ordnance.	
nel Shell.		Palliser Shot complete with Gas-check, and weighted up with sand.			Bursting Charges.									
ith Gas-					Common Shell.				Shrapnel Shell R.F.G., F.G., or Pistol.					
					L.G.		P. & F.G.						Guns.	Mark.
lb.	oz.	Mark.	lb.	oz.	lb.	oz.	lb.	oz.	lb.	oz.	M			
1995	0	I	2000	0	78	0	106	0	5	0		V	17·72-in. (100 tons)	I
1695	13	II	1700	0	60	0	75	6	4	3		M	16-in. (80 tons)...	I
815	9	III	818	0	33	0	37	8	2	7		Long, solid drawn & electric	12·5-in. (38 tons)	I
815	9	III	818	0	33	0	37	8	2	7		V	12·5-in. (38 tons)	II
711	15½	II	714	0	27	4	34	10	2	0½		Long, solid drawn & electric	12-in., 35-ton ...	I
612	0	III	614	0	23	6	28	2	2	0		,,	12-in., 25-ton ...	II
545	15½	III	548	0	22	0	27	0	2	0½		,,	11-in. (25 tons,	II
460	4½	I	462	0	...		24	9	1	9		P	10-4-in. (28 tons)	I
408	6	III	410	0	19	9	23	8	1	10		Long, solid drawn & electric	10-in. (18 tons)	I
253	14	III	256	0	14 ...	8	17 27	0 12	1	5		,,	9-in. (12 tons) ...	V
					14	8	17	0				,,	9-in (12 tons) ...	VI

MUZZLE-LOADING ORDNANCE (R.M.L.).—(*Continued*).

Ordnance.		Material	Nature of Carriage (giving length of Recoil, size of Port, &c., where necessary).	Elevation in Degrees.	Depression in Degrees.	Height of Axis of Trunnions in inches.	Radii of Racers in feet and inches.		Weight in cwt.	
Nature of Guns.	Weight						Front.	Rear.	Carriage (without Gun).	Limbers (without Stores).
	Tons.						ft. in.	ft. in.		
17·72-in.	100	Iron	... ...	11	11	52	...	...	407·06	...
16-in.	80	,,	... ...	7	2	44	...	...	...	...
12·5-in.	38	,,	7-ft. recoil, small port ...	7	4	A 32·9 B 46·9	...	...	206·75	...
			6-ft. ,, ,, ...	7	4	A 32·9 B 46·9	...	...	193·75	...
			7-ft. ,, ...	10	5	36·65	...	...	123·5	...
			6-ft. ,, ...	10	5	36·65	...	...	122·25	...
12-in.	35	,,	... ...	10	5	36·65	...	...	115·25	...
	25	,,	... ...	15	5	41	...	...	67·75	...
11-in.	25	,,	... ...	15	5	41	...	...	67·75	...
10·4-in.	28	,,	... ...	15	4½	38·5	...	...	121	...
10-in.	18	,,	Small port	9	4	A 32·9 B 44·9	...	...	120	...
			Casemate, Mark II.... ...	10	5	23	...	...	67	...
			,, or Dwarf... ...	10	5*	33·5	...	...	51·25	...
9-in.	12	,,	,, ,, ...	10	5	31·5	...	..	42·25	...
			Barbette, 35°	85	12	31·5	...	...	48	...
			Converted Naval S.P. ..	12	9	31·5	...	...	44	...
			Moncrieff, with platform...	15	5	168	11 10	6 2	802·5	...
8-in.	9	,,	Converted Naval S.P. ...	12	9	28	...	...	39·25	...
			Elevation	16	9	28·25	..	...	39	...
7-in.	7	,,	Casement or dwarf	20	5	31·25	...	...	27·75	...
	6½	,,	Sliding medium, No. 7, 6-ft. parapet	30	5	48	...	...	54·25	...
			Converted Naval S.P., with hydraulic buffer	15	7	31·062	...	...	32	...
			Converted Naval S.P., with compressor gear	15	7	30·5	...	...	27·25	...
			Converted Naval S.P., with hydraulic buffer	13·5	7	30·5	...	...	27·5	...
	7	,,	Moncrieff, with platform (Mark I.)	15	5	131·75	8 3·5	5 8·5	443·5	...
			Moncrieff, with platform (Mark II.)	15	5	147·5	9 10	5 7·5	438	...
6·6-in....	Cwt. 570	Steel	Siege (H.P.)	12	5	101·5	...	...	52·4	11·87
8 0-pr. (convd.)...	Tons. 5	Wood	Sliding medium, No. 21, casemate	7 F	5	29·25	...	...	15·25	...
			Sliding medium, No. 22, dwarf	18 F	5	36	...	...	15·75	...
64-pr.	Cwt. 64	Iron	Sliding medium, No. 8, 6 ft. parapet	15	5	47·5	...	...	34·75	...
			Sliding medium, No. 9, 5 ft. 6 in. parapet	15	5	42	...	...	27	...
			Sliding medium, No. 26, 6 ft. parapet, with hydraulic buffer	15	6	47·5	...	...	34·75	...

MUZZLE-LOADING ORDNANCE (R.M.L.).—(*Continued*).

Ordnance.		Material	Nature of Carriage (giving length of Recoil, size of Port, &c., where necessary).	Elevation in Degrees.	Depression in Degrees.	Height of Axis of Trunnions in inches.	Radii of Racers in feet and Inches.		Weight in cwt.	
Nature of Guns.	Weight						Front.	Rear.	Carriage (without Gun).	Limbers (without Stores).
64-pr. (convd.)	Cwt. 71 & 58	Wood	Common standing	17 F	6	40·5	...	...	14·75	...
	71 & 58	Iron	,, ...	20	5	41	...	...	17·25	..
	71	Wood	§ Sliding medium, No. 23	20 F	5	35	...	...	12·5	...
		Iron	,, No. 10, 4-ft. 3-in., or 3ft. 6-in. parapet	15	5	36	...	...	23	...
			Sliding medium, No. 11, 3-ft. 6-in. parapet	15	8	36	...	...	24·5	...
			Sliding medium, No. 12, 2-ft. 7-in. parapet	12	8·5	33	...	...	23	...
	58	Wood	‡ Sliding medium, No. 24, 6-ft. parapet	15 F	6	34	...	...	14	...
			Depression	...	30	40·5	...	...	16	...
			Sliding medium, No. 25, casemate	10 F	6	29	...	...	11	...
		Iron	** Sliding medium, No. 13, 4-ft. 3-in., or 3-ft. 6-in. parapet	15 ** 13·5 ††	5** 8·5 ††	36	...	...	22·25	...
			Depression (Mark I.) ...	...	32	42·875	...	...	29·25	...
			Moncrieff, with platform	10·5	5	125	C 6 10 D 8 11	C 6 10 D 4 8·5	229	...
40-pr.	34 & 35	Iron	Sliding medium, No. 14 ...	10	8	23	...	...	11·5	...
			Siege, Mark I.	35	5	53	...	...	27·05	13·25
			,, Mark II.	35	5	53	...	...	32·5	13·25
			Top, overbank...	35	5	75	...	...	10·5	...
25-pr.	18	,,	Field...	45	5	46	...	...	15	11·25
			Siege, top overbank... ...	35	5	72	...	...	8·5	...
16-pr.	12	,,	Mark I., strengthened ...	22·5	12·25	43·25	...	...	13·42	11·33
			,, II.	17	15	43·25	...	...	13	11·17
15-pr. (jointed) ...	Lbs. 422	Steel	Mountain, Mark I.	25	10	25·6	...	...	3·75	..
13-pr.	Cwt. 8	,,	Field...	16	5	43	...	...	12	12
9-pr.	8 & 6	Iron	Mark I., strengthened ..	21	4	42·5	...	...	12·82	11·43
			Mark II.	22	6	42·5	...	...	11·82	11·34
2·5-in. (jointed)	Lbs. 400	Steel	Mountain, Mark II.... ...	25	10	25·75	...	...	4·75	...
7-pr.	200	Iron	Field...	35	10	44·75	...	...	9·25	9·25
			Mountain	33	8	25·75	...	...	3·125	3·41
	150	Iron	,, ...	45	5	22	...	...	2·75	-
		Steel	Gold coast	20	5	24	...	...	3·187	...
		Wood	,, ...	20	5	24	...	...	2·5	...
Howitzer Carriages.										
8-in.	Cwt. 70	Iron	70-cwt., siege	35	5	53	...	..	44 D	12·1
	46	,,	46-cwt., ,, ...	30	...	56·5	...	...	51·5D	12·1

MUZZLE-LOADING ORDNANCE (R.M.L.).—(*Continued*).

Ordnance.		Material	Nature of Carriage (giving length of Recoil, size of Port, &c., where necessary).	Elevation in Degrees.	Depression in Degrees.	Height of Axis of Trunnions in inches.	Radii of Racers in feet and inches.		Weight in cwt.	
Nature of Guns.	Weight						Front.	Rear.	Carriage (without Gun).	Limbers (without Stores).
	Cwt.									
6·6-in.	36	Iron	Siege...	35	5	53	...	...	38 D	12·1 E
6·3-in....	18	,,	,, ...	40	5	53	...	...	37·5D	12·1 E
	Lbs.									
4-in. (jointed) ...	600	Steel	Mountain, jointed	36	7	26·875	...	...	5·37	...
Beds.—Guns.										
	Lbs.									
7-pr.	200	Iron	7-pr., 200-lb.	22	5	16·25	...	...	2	...
Howitzers.										
	Cwt.									
8-in.	46	,,	8-in. 46-cwt.	45	...	30	...	...	33	...
6·6-in....	36	Steel	6·6-in.	45	...	36·5	...	...	31	...
6·3-in....	18	,,	6·3-in.	70	...	36·5	...	...	31·25	...

* When mounted on slide 7-ft. parapet "C."
† Also for 68-pr. S.B.
‡ Also for 32-pr. S.B. of 58 or 56-cwt.
** For 4-ft. 3-in. parapet.
†† For 3-ft. 6-in. parapet.
§ Also for 8-in. S.B. of 65 or 60-cwt.

A. Least height.
B. Greatest height.
C. "C" pivot.
D. "D" pivot.
E. For siege-train 12·5-cwt.
F. Elevation from bed, screw removed.

Ordnance.		Slides.								
Guns.	Weight	Material	Nature of Slide (whether Casemate, Dwarf, &c.).	Height at Axis of Trunnions.	Radii of Racers.		Radius of Rack.	Radius of Traversing Arc.	Weight of Slide.	Nature of Carriage or Carriages used, with Slide.
					Front.	Rear.				
	Tons.			in.	ft. in.	ft. in.	ft. in.	ft. in.	cwt.	
17·72-in....	100	Iron	... ...	66	...	...	...	...	514·75	Garrison, iron sliding
16-in. ...	80	,,	... ...	33·25	...	...	...	...	...	Turret L ... ...
12·5-in. ...	38	,,	Casemate, 7-ft. recoil ...	21·6	10 2	21 2	15 5	19 4½	163¼	7-ft. recoil ...
			,, 6-ft. ,, ...	22·6	10 2	20 2	14 5	19 4½	162¼	6-ft. ,, ...
			,, 6-ft. ,, special	22·6	10 2	21 2	14 5	19 4½	168	6-ft. ,, ...
			Dwarf, C pivot..	35·6	5 8	5 8	3 4·47	4 9¾	188¼	6-ft. ,, ...
			,, D ,, ...	35·6	8 0	3 8	5 6	6 9	184½	6-ft. ,, ...
			Small port, 7-ft. recoil ...	22·6	10 2	21 2	15 5	19 4½	150	Small port, 7-ft. recoil
			,, 6-ft. ,, ...	22·6	10 2	20 2	14 5	19 4½	169½	Small port, 6-ft. recoil
12-in. ...	35	,,	Dwarf, C pivot...	35·6	5 8	5 8	3 4·47	4 9¼	186¾	6-ft. recoil ...
12-in. 11-in.	25	,,	Casemate to work within length	18·5	8 0	18 0	...	21 4¾	112	Casemate or dwarf
			Dwarf, A pivot, rear ...	34	8 0	18 0	...	21 4¾	139¼	
			,, C ,, central ...	34	5 8	5 8	...	7 9	142½	
			7-ft. parapet, C pivot ...	34	5 8	5 8	...	5 10	...	

MUZZLE-LOADING ORDNANCE (R.M.L.).—(*Continued*).

Ordnance.		Slides.								
Guns.	Weight	Material	Nature of Slide (whether Casemate, Dwarf, &c.).	Height of Axis of Trunnions.	Radii of Racers. Front.	Radii of Racers. Rear.	Radius of Rack.	Radius of Traversing Arc.	Weight of Slide.	Nature of Carriage or Carriages used, with Slide.
	Tons.			in.	ft. in.	ft. in.	ft. in.	ft. in.	cwt.	
10·4-in. ...	28	Iron	Barbette	71	...	12 10·8	...	...	148	Barbette
10-in. ...	18	,,	Casemate to work within length, Mark I.	17	8 0	18 0	...	21 4¾	97½	Casemate or dwarf
			Casemate to work within length, Mark II.	27·5	8 0	18 0	...	21 4¾	104	Casemate Mark II.
			Casemate to work within length, Mark II., special	26	8 0	18 0	...	21 4¾	125	
			Casemate to work within length, Mark II., convd.	26	8 0	18 0	...	21 4¾	139½	
			Dwarf, A pivot, rear ...	38·5	8 0	18 0	...	21 4¾	134¼	Casemate or dwarf
			,, C ,, central ...	38·5	5 8	5 8	...	7 9	140½	
			,, D ,, ...	38·5	9 0	3 0	...	3 8¾	141	
			7-ft. parapet, C pivot ...	38·5	5 8	5 8	...	7 9	159¼	
			Small port	22·6	8 0	18 0	...	21 4¾	106	Small port
9-in. ...	12	,,	Casemate to work within length	18	6 3	16 6	...	20 0	74¾	Casemate or dwarf
			Dwarf, A pivot, rear ...	37·5	6 3	16 6	...	20 0	99	
			,, C ,, central ...	37·5	5 5¾	5 5¾	...	5 10	110	
			,, D ,, ...	37·5	9 0	2 3¾	...	5 10	107	
			7-ft. parapet, C pivot ...	37·5	5 5¾	5 5¾	...	5 10	121	
			Converted (Naval) slide ...	17·5	5 1	13 6	11 5½	...	60	Convd. Naval S. Plate
			C pivot, 35° Barbette, Mark I.	37·5	5 5¾	5 5¾	...	5 10	130	Barbette 35°
			C pivot, 35° Barbette, Mark II.	37·5	5 5¾	5 5¾	...	5 10	142	
8-in. ...	9	,,	Converted (Naval) slide ...	17·5	5 1	13 6	11 5½	...	55.25	Convd. Naval S. Plate
			Elevation	13	4 7½	13 7	10 5½	...	55·5	"Elevation"
7-in.	7	,,	Casemate	18	6 3	16 6	...	...	53	Casemate or dwarf
			Dwarf, A pivot...	37·5	6 3	13 6	...	...	77¾	
			,, C ,, ...	37·5	5 5¾	5 5¾	...	...	80¾	
			,, D ,, ...	37·5	9 0	2 3¾	...	...	80¾	
	6½	,,	Dwarf, S.P. carriages C pivot	37·5	5 5¾	5 5¾	...	...	80¾	Convd. Naval S. Plate
			Dwarf, S.P. carriages D pivot	37·5	9 0	2 3¾	...	...	80¾	
			Converted slide, 4° slope, for hydraulic buffer	10	5 1	13 6	...	...	39	Convd. Naval, with hydraulic buffer
			Converted slide, 4° slope, with compressor gear	9·75	5 1	13 8	...	...	37½	Convd. Naval, with compressor gear
			Traversing medium, No. 7, 6-ft. parapet	37	6 1	6 1	...	...	75½	Iron sliding medium, No. 7
80-pr. (convd).	5	Wood	Traversing medium, No. 14, casemate	13·5	5 0	16 6	...	...	27	Wood sliding medium, Nos. 15, 19, 21 and 25
			Traversing medium, No. 16, dwarf	26	C 6 1 D 9 0 E 10 8¼ F 12 10	C 6 1 D 3 4½ E 2 2 F 2 2	...	...	37	Ditto, Nos. 16, 17, 22 and 23
			Traversing medium, No. 18, 6-ft. parapet	48	6 1	6 1	...	...	53	Ditto, Nos. 22, 23 and 24

MUZZLE-LOADING ORDNANCE (R.M.L.).—(*Continued*).

Ordnance.		Slides.								
Guns.	Weight	Material	Nature of Slide (whether Casemate, Dwarf, &c.).	Height at Axis of Trunnions.	Radii of Racers.		Radius of Rack.	Radius of Traversing Arc.	Weight of Slide.	Nature of Carriage or Carriages used, with Slide.
					Front.	Rear.				
				in.	ft. in.	ft. in.	ft. in.	ft. in.	cwt.	
64-pr. ...	Cwt. 64	Iron	Traversing medium, No. 8, 6-ft. parapet	36·5	A 5 0 C 6 1 E 10 8¼ F 12 10	A 16 6 C 6 1 E 2 2 F 2 2	...	...	69¾	Iron sliding medium, No. 8
			Traversing medium, No. 9, 5-ft. 6-in. parapet	36	C 6 1 D 9 0 E 10 8¼ F 12 10	C 6 1 D 3 4½ E 2 2 F 2 2	...	...	58	Ditto No. 9
			Traversing medium, No. 21, 6-ft. parapet	36	C 6 1 D 9 0 E 10 8¼ F 12 10	C 6 1 D 3 4½ E 2 2 F 2 2	...	...	...	Ditto No. 26
64-pr. (convd.)	71	Wood	Traversing medium, No. 13 (11-ft. long), for 3-ft. 6-in. parapet	18	5 0	14 0	...	...	30¾	Ditto Nos. 3 & 11
			Traversing medium, No. 19, 2-ft. 7-in. parapet	13	5 0	16 6	...	...	33¼	Ditto No. 12
			Traversing medium, No. 20, 6-ft. parapet	48	5 0	16 6	...	...	63	Ditto No. 10
	71 & 58	,,	Traversing medium, No. 11, 4-ft. 3-in. parapet	25·75	C 6 1 D 9 0 E 10 8¼ F 12 10	C 6 1 D 3 4½ E 2 2 F 2 2	...	...	44	Ditto Nos. 2, 10 and 13
			Traversing medium, No. 12, 3-ft. 6-in. parapet	18·5	5 0	16 6	...	...	34¾	Ditto Nos. 2, 10 and 13
			Traversing medium, No. 16, dwarf	26	C 6 1 D 9 0 E 10 8¼ F 12 10	C 6 1 D 3 4½ E 2 2 F 2 2	...	...	37	Wood sliding medium, Nos. 16, 17, 22 and 23
			Traversing medium, No. 18, 6-ft. parapet	48	6 1	6 1	...	...	53	Ditto Nos. 22, 23 and 24
	58	,,	Traversing medium, No. 14, casemate	13·5	5 0	16 6	...	...	27	Ditto Nos. 15, 19, 21, and 25
40-pr. ...	...	Iron	Traversing medium, No. 10	19	5 0	14 0	...	...	24½	Iron sliding medium, No. 14

A, C, D, E, F, represent pivots of corresponding letter.

3.—QUICK-FIRING AND

QUICK-FIRING

Ordnance, Quick-firing.						Bore.		Ballistic Effects.			
										Penetration of Steel Plate in Inches—	
Nature.	Mark.	Weight.	Material.	Service.	Total Length in Inches.	Calibre in Inches.	Length in Inches.	Muzzle Velocity in foot-seconds.	Muzzle Energy in foot-tons.	At Muzzle.	At 1,000 yds.
4·7-inch	I II	41 cwt.	Steel	I S.S. II {L.S. S.S.}	194·1	4·724	189	1786	995	6·7	5·4
Hotchkiss, 6-pr. ...	I II*a*	8 cwt.	Steel	I {S.S. and L.S.} II L.S.	97·63	2·244	89·76	1818	136·9	3·2	2·0
Nordenfelt, 6-pr. ...	I II III*a*	6 cwt.	Steel	I S.S. II {S.S. and L S.} III L.S.	104·4	2·244	95	1837	141·2	3·3	2·1
Hotchkiss, 3-pr. ...	I II*a*	5 cwt.	Steel	I {L.S. S.S.} II L.S.	80·63	1·85	74 06	1873	80·2	2·9	1·8
Nordenfelt, 3 pr. ...	I	4 cwt.	Steel	L.S.	91·5	1·85	84·0	*e*	*e*	*e*	*e*

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a. As regards the construction of the gun itself, the 6-pr. Hotchkiss Mark II, the 3-pr. Hotchkiss Mark II, and the 6-pr. Nordenfelt Mark III, are exactly similar to the 6-pr. Hotchkiss Mark I, the 3-pr. Hotchkiss Mark I, and the 6-pr. Nordenfelt Mark II, respectively. The differences constituting the latter Mark in the case of the 6-pr. guns are the removal of the shoulder-piece and trigger-guard with trigger from the gun to the ("recoil") mounting; and in the case of the 3-pr. guns, the guns of later Mark being fitted for field carriages. The 6-pr. Hotchkiss Mark I, the 3-pr. Hotchkiss Mark I, and the 6-pr. Nordenfelt Mark II, have both shoulder-piece and trigger-guard with trigger on the gun, the mounting being "non-recoil." In the Naval Service, however, the 3-pr. Hotchkiss Mark I is sometimes mounted on a recoil mounting, in which case the shoulder-piece is removed from the gun. The gun on this mounting is fired by a lanyard attached to the trigger, and not directly by hand, as in the 6-pr. recoil mounting.

e. Not yet ascertained.

MACHINE GUNS.

GUNS.

Rifling.						Fuzes.		Ordnance, Q.F.	
System.	Twist.	Length in Inches.	Grooves.			Time.	Percussion.	Nature.	Mark.
			Number.	Width in Inches.	Depth in Inches.				
Polygroove E.O.C.	I. from 1 in 100 at breech, to 1 in 34·352 at 6·65 inches from the muzzle. Remainder U. 1 in 34·352	171	22	0·5	0·04	Time & Percussion Middle *c* Middle sensitive *d*	Base Percussion, Armstrong	4·7-inch	I II
Polygroove (plain section)	I. from 1 in 180 at breech, to 1 in 29·89 at 9·98 inches from the muzzle. Remainder U. 1 in 29·89	76·91	24	0·22	0·012	—	Base Percussion, Hotchkiss or Nordenfelt	Hotchkiss, 6-pr.	I II
Polygroove (plain section)	I. from 1 in 180 at breech, to 1 in 29·89 at 14·98 inches from the muzzle. Remainder U. 1 in 29·89	81·91	24	0·22	0·012	—		Nordenfelt, 6-pr.	I II III
Polygroove (plain section)	U. 1 in 25 ..	58·35	20	0·23	0·015	—		Hotchkiss, 3-pr.	I II
Polygroove (plain section)	I. from 1 in 100 at breech, to 1 in 30 at 9·7 inches from the muzzle. Remainder U. 1 in 30	68·217	20	0·194	0·012	—		Nordenfelt, 3-pr.	I

c. For present use.

d. For future use.

Machine Guns. — Nature.	Calibre.	Number of Barrels.	Mark and Letters.	Service.	Weight of Gun complete, with Distributor or Cartridge Feeder, but without Mounting or Shield.	Total Length.	Sighted up to	Length of Barrels.	Rifling.			
									System.	Pitch.	Length.	Number of Grooves.
	in.				lb.	in.	yards.			in.	in.	
Nordenfelt	1·0	2	I	S	180	52·75	3000	35·48	Henry	35	31·39	11
,, ...	1·0	4	I	S	440	57·0	3000	35·5	,,	60	31·5	11
,, ...	1·0	4	II	S	440	57·0	3000	35·5	,,	35	31·5	11
,, ...	0·45	4	III	S	417	57·0	3000	35·48	,,	35	31·37	11
,, ...	0·45	3	M.H.	L	103	41·5	1800	28·5	,,	22	25·6	7
,, ...	0·45	5	I G.G.	S	164	46·0	2000	28·5	,,	22	25·6	7
,, ...	0·45	5	II G.G.	L & S	143	42·25	2000	28·5	,,	22	25·6	7
Gardner	0·45	1	G.G.	S	76	47·0	2000	30·0	,,	22	27·1	7
,, ...	0·45	2	G.G.	L & S	218	47·0	2000	30·0	,,	22	27·1	7
,, ...	0·45	2	I M.H.	L	218	47·0	2000	30·0	,,	22	27·1	7
,, ...	0·45	5	I G.G.	L & S	290	53·5	2000	33·0	,,	22	30·1	7
,, ...	0·4	2	I E.M.	L	88	45·5	2000	28·5	Enfield	15	25·5	7
Gatling	0·65	10	—	S	817	66·5	2000	33·0	Henry	30	28·145	7
,, ...	0·45	10	G.G.	L & S	444	59·41	2000	31·95	,,	22	28·79	7
,, "Accles' Feed"	0·45	10	G.G.	S	266	51·0	2000	32·0	,,	22	29·1	7
Maxim	0·45	1	G.G.	S	63	43·6	2000	29·2	,,	22	26·0	7
,, ...	0·45	1	M.H.	L	60	43·6	2000	29·2	,,	22	26·0	7
,, ...	0·303*	1	—		—	—	—	—	—	—	—	—

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* A gun of this calibre has been approved, but the exact dimensions, weight, &c., are as yet undecided.

Weight of Hopper, Drum Cartridgeholder, Feed-box, or Belt.		Number of Cartridges in Hopper, Drum, Cartidgeholder, Feed-Box, or Belt.	Nature of Ammunition.	Powder Charge.	Projectile.	Penetration.	Nature of Gun.
Empty.	Filled.						
lb.	lb.			grains.			
13	27	20	Solid case, steel projectile	625 M.G.[1]	7·25 oz.	Perforates ¾" steel plate at 200 yards	Nordenfelt, 1·0-in., 2-bl., Mark I.
20	48	40	Solid case, steel projectile	625 M.G.[1]	7·25 oz.	Perforates ¾" steel plate at 200 yards	Nordenfelt, 1·0-in., 4-bl., Mark I.
20	48	40	Solid case, steel projectile	625 M.G.[1]	7·25 oz.	Perforates ¾" steel plate at 200 yards	Nordenfelt, 1·0-in., 4-bl., Mark II.
20	48	40	Solid case, steel projectile	625 M.G.[1]	7·25 oz.	Perforates ¾" steel plate at 200 yards	Nordenfelt, 1·0-in., 4-bl., Mark III.
0·7½	3·12	27	M.H., solid case	85 R.F.G.[2]	480 gr.	Same as M.H. Rifle, which perforates ⅛-inch wrought-iron plate at 600 yards, 3/16-inch plate at 400 yards, and ¼ in. at 100 yards	Nordenfelt, ·45-in., 3-bl., M.H.
5·2	10·14	50	Gardner-Gatling	85 R.F.G.[2]	480 gr.		Nordenfelt, ·45-in., 5-bl., Mark I.
5·2	10·14	50	,, ,,	85 R.F.G.[2]	480 gr.		Nordenfelt, ·45-in., 5-bl., Mark II.
7/16	2⅜	20	,, ,,	85 R.F.G.[2]	480 gr.		Gardner, ·45-in., 1-bl., G.G.
7/16	3⅜	30	,, ,,	85 R.F.G.[2]	480 gr.		Gardner, ·45-in., 2-bl., G.G.
7/16	2·14½	30	M.H., solid case	85 R.F.G.[2]	480 gr.		Gardner, ·45-in., 2-bl., M.H.
3 5/16	9·	100	Gardner-Gatling	85 R.F.G.[2]	480 gr.		Gardner, ·45-in., 5-bl., G.G.
7/16	2 13/32	20	·4-inch, solid case	85 R.F.G.[2]	384 gr.	Not known	Gardner, ·4-in., 2-bl.
28·6	46·4	50	·65-inch, rolled case	270 R.F.G.[2]	1422 gr.	,,	Gatling, ·65-in.
29·	56·12	240	Gardner-Gatling	85 R.F.G.[2]	480 gr.	Same as M.H. Rifle	,, ·45-in., G.G.
17½	29 1/16	100	,, ,,	85 R.F.G.[2]	480 gr.	,, ,,	,, ·45-in., G.G. (Accles').
Not known			,, ,,	85 R.F.G.[2]	480 gr.	,, ,,	Maxim, ·45-in., G.G.
2½	34	250	M.H., solid case	85 R.F.G.[2]	480 gr.	,, ,,	,, ,, M.H.
—	—	—	—	71½	215 gr.	Not known	,, ·303-in.

Ordnance, Q.F.		Projectiles and their Bursting Charges.								
Nature.	Mark.	Mark.	Steel Shell, Weight, filled, and fuzed.	Bursting Charge, Steel Shell.	Mark.	Iron Shell, Weight, filled and fuzed, or plugged and filled with salt. (Practice). *b.*	Bursting Charge, Iron Shell.	Mark.	Shrapnel Shell, Weight, filled and fuzed.	Bursting Charge, Shrapnel Shell.
			lb. oz. dr.	lb. oz. dr.		lb. oz. dr.	lb. oz. dr.		lb. oz. dr.	lb. oz. dr.
4·7-inch	I II	III	45 0 0	1 15 0 P.&F.G.	II	45 0 0	2 15 8 P.&F.G.	I	45 0 0	0 5 0
Hotchkiss, 6-pr. ...	I II	III	6 0 0	0 4 0 F.G.	II	6 0 0	0 3 0 F.G.	—	—	—
Nordenfelt, 6-pr....	I II III									
Hotchkiss, 3-pr. ...	I II	II	3 4 12¾	0 2 0 F.G.	II	3 4 5¼	0 1 6 F.G.	—	—	—
Nordenfelt, 3-pr....	I									

b. The 6-pr. and 3-pr. iron shell are used for practice only, and are filled with salt and plugged for naval service. For practice from land-service guns, either plugged or filled shell may be used.

GUNS.

Description of Ammunition.	Charges.		Carriages.									Ordnance, Q.F.	
	Service.	Saluting.	Nature of Carriage.	Elevation in Degrees.		Depression in Degrees.		Height of Axis of Trunnions in Inches.		Weight of Carriage, without Gun.		Nature.	Mark.
				Embrasure.	Cone.	Embrasure.	Cone.	Embrasure.	Cone.	Embrasure.	Cone.		
Charge in brass solid drawn case, with axial primer (electric) attached. Projectile separate	12lbs. S.P.	3lbs.	—	—	—	—	—	—	—	—	—	4·7-inch ...	I II
"Fixed" Ammunition, central fire, viz., Charge in brass solid drawn case, with percussion cap in base, and projectile attached, as with Small-Arm Ammunition	1lb. 15ozs. Q.F.[1]	15ozs.	Q.F. Recoil	20°	20°	20°	20°	22·55 above sill	43·435	7·43 with shield	7·65 with shield	Hotchkiss, 6-pr.	I II
			Q.F. Recoil	20°	20°	20°	20°	22·55 above sill	43·435	7·08 with shield	7·3 with shield	Nordenfelt, 6-pr.	I II III
	1lb. 8ozs. Q.F.[1]	15ozs.	Field	12°		5° 30′		41·5		cwt. qr. lb. 11 3 0 (*c*)		Hotchkiss, 3-pr.	II
			,,	15°		5° 30′		41·5		10 2 21 (*d*)		Nordenfelt, 3-pr.	I

c = Weight of carriage. *d* = Weight of limber.